MACHINE DESIGN

August 1943

In This Issue:

Design Peers Ahead!

Ceramics in Machines

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Contents

For Itemized Table of Contents See Page 5

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Itemized Index

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Topics

EXPLODED metal bubbles to facilitate bonding on hard metal surfaces, especially those which cannot be roughened by grit blasting or cutting methods, are being produced by sparking a nickel alloy electrode over the surface. The electrode heats up by its own resistance and explodes into a foam like frozen soap bubbles fused to the base metal. The resulting rough, porous surface is ideal for bonding a sprayed metal coating.

AIR CONDITIONING for submarines—both cooling and reversed refrigeration cycles together with air-purifying chemicals—is making life more pleasant for the crew. Members can even smoke aboard, a privilege unheard of during World War I. During arctic tours the refrigerating system may be run backward to heat the craft.

PRODUCTION of synthetic rubber has been expedited by locating three plants side by side—a Dow styrene, a Shell butadiene and a Goodyear-Firestone synthetic crepe rubber plant. Styrene and butadiene are pumped from their respective plants to the "crude" plant effecting a continuous process between the three.

STANDARDS for truncated Whitworth screw threads are being established by the recently formed War committee organized by the American Standards association at the request of the War Production Board to remove difficulties experienced in this country in the manufacture and maintenance of taps, dies and gages of British-made Whitworth threads.

SYNTHETIC JEWEL, known as "spinel", has been classified as a jewel bearing material by WPB. Suitable for use in delicate instruments or where sapphires have been used as bearings, spinel under this ruling ceases to be known as a "substitute" as formerly.

PORTABLE sectional pipe lines, readily movable for installation over difficult terrain to carry gasoline supplies to motorized units in the field, have been developed by the Army. Complete with centrifugal pump and 20-horsepower gasoline engine, each section is a self-contained unit and capable of handling 200 gallons per minute.

ZEBRA MARKINGS of black and white stripes are utilized on the new Hawker Typhoon, Britain's most formidable single - engined fighter, because of the resemblance between it and the new Focke-Wulf Fw190. The markings on the under surface of the wings reduce the risk of the fighters being mistaken by ground gunners.

SMALL REPEATER TUBES are being developed by Bell laboratories that may be incorporated within a cable and treated as part of it when reeled off the drum. This development, a pentode tube with an indirectly heated nickel cathode, will make transoceanic telephone cables a practical possibility. Telephone calls now are really transmitted across the ocean by radio.

AMERIPOL derivative compound has produced a synthetic rubber compound that will remain flexible at —70 degrees Fahr. and can be bent at an angle of 90 degrees around a ½-inch rod. The new compound is being used for sealing strips in bolted tanks for storage of high octane gas and aromatic fuels used in military aircraft.

COMPRESSOR HORSEPOWER built by the United States since Pearl Harbor is believed by authorities to have an important bearing on our enemies' reluctance to carry out their poison-gas threats. Compressors built for common gases can be diverted easily to poison-gas production.

GOODS which went into homes the year before the war used the following proportions of the total year's output of materials: Aluminum, 22 per cent; copper and brass, 19 per cent each; zinc, 20 per cent; tin, 23 per cent; steel, 30 per cent; nickel, 16 per cent and chromium, 15 per cent. Now these are replaced largely by substitutes in available consumer goods.

The Shape of Things To Come

Part I of a series of three articles dealing with design of the future from the appearance standpoint

By R. S. Elberty
Consulting Engineer

IN TIME of war, does everybody lose? Did not the greatest industrial era in the development of our nation follow the World War? Let us hope that some such benefit will follow our present struggles and that the progress made in destruction will help toward the reconstruction of a still better world. Surely the incentive for intensive engineering ing development has never been greater. This war of engineering has brought new materials, new methods and a new approach to machine design. Economics of the use of materials in manufacturing processes has compelled a functional type of design that is sound because of its basic simplicity. Function replaces ornament "for the duration" and, let us also hope, for the next decade of machine design.

Just as modern industrial machinery has been influenced by the styling of automobiles and household appliances, these very items of the postwar era may well utilize soundness of design found in machine tools and tanks. The design period for the coming world is now at hand. Engineers are thinking ahead, planning to use our new riches in materials and techniques for the improvement of the machines for peacetime living.

Machine designers have always been style conscious. There have been good and bad designs, but they have been designs—the attempt had been made. The designer can however, utilize a knowledge of basic design principles to *plan* his machine, knowing that the result will be better than a haphazard grouping of elements. The word "symmetry" used in the sense of analogy signifies the relationship which the composing elements of form in design (or in natural growth) bear to the whole. In design it is that quality which governs the just balance of variety to effect unity in appearance, illustrated aptly in *Figs.* 1 and 9.

There are two types of symmetry found in natural growth, arithmetic or static and geometric or dynamic. In the field of art, static symmetry is the obvious and therefore the common basis of design. Chinese, Japanese, Persian, Hindu, Byzantine, and Gothic art all belong to the static classification of symmetry. A different classification of design started with the Egyptians and reached

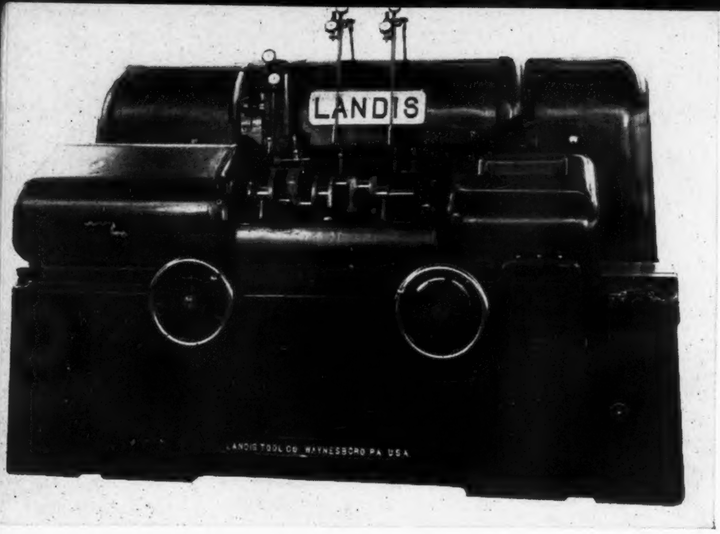


Fig. 1—Grinder design employs the principles of dynamic symmetry. It embodies the same proportions for headstock and other parts to integrate appearance

a high stage of perfection in Greek art. This superior design basis is geometrical in nature and is called "dynamic symmetry". The Greeks excelled in the fields of art and geometry and they combined the two to good effect. It is no coincidence that the Parthenon is dimensioned with true geometrical ratios, the errors being less than one per cent. The principles of dynamic symmetry are simple and easily grasped by the engineering or mathematical mind, but it is small wonder that artists themselves have difficulty in understanding these values.

There are points in common between static and dynamic symmetry, in fact static symmetry has been described as a special limited case of dynamic symmetry. For example, a square is a special form of rectangle, both arithmetical and geometrical in nature. So a square may be selected for the starting point, a foundation for the study of dynamic symmetry, *Fig.* 2 being selected as a unit square. metrical construction is shown in *Fig.* 3.

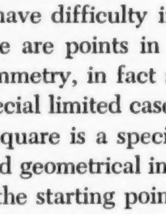


Fig. 2 — Unit square which is the basis of dynamic symmetry

Another square erected on the diagonal of the unity square will have an area of two, the length of one side being $\sqrt{2}$. A rectangle can now be constructed, the short side $= 1$, the long side $= \sqrt{2}$. This simple-root rectangle is called the "root 2 rectangle" and its geo-

A square erected on the diagonal of a root 2 rectangle will have an area of $(1)^2+(\sqrt{2})^2=3$ and one side will be equal to $\sqrt{3}$. A rectangle, one side of which $= 1$ and the long side of which $= \sqrt{3}$, will be the root 3 rectangle. The construction is shown in *Fig.* 4. And so a root 4 rectangle may be developed using similar construction. However a root 4 rectangle is also an arithmetic rectangle because it can be constructed by adding 2 squares, since $\sqrt{4}=2$. The root 5 rectangle may be constructed from a root 4 as shown in *Fig.* 5.

Both the root 4 and the root 5 rectangles are important, the root 4 because of its close relationship to static symmetry, and the root 5 because of its relationship in the complex rectangles to be discussed later. There is no reason why the system cannot be expanded, but simple-root-rectangles longer than the root 5 are usually not considered. Perhaps this is because longer rectangles are harder to estimate by eye. Longer shapes, however, are necessary in machine design and there is nothing to prevent the designer from using root 6, and root 7 and longer rectangles if he so desires.

If the longest side of a rectangle is considered as unity,

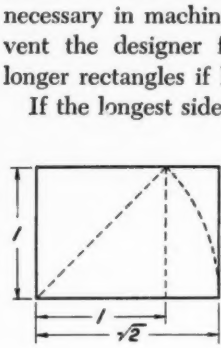


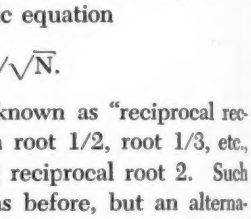
Fig. 3—Simple-root rectangle having diagonal of basic square for length of side

the short side becomes less than unity. These rectangles, however, are similar to the root rectangles and the sides are in mean and extreme proportion which may be expressed generally in the algebraic equation

$$\sqrt{N}:1=1:1/\sqrt{N}.$$

These smaller rectangles are known as "reciprocal rectangles" which might be written root 1/2, root 1/3, etc., the expression root 1/2 meaning reciprocal root 2. Such rectangles may be constructed as before, but an alternative method of construction will show many interesting relationships. Referring to *Fig.* 6, an arc centered on one

Fig. 4—Root 3 rectangle having diagonal of Fig. 2 for one side



corner of the square is described in the square as shown. At the intersection of this arc with the diagonal, a line is drawn perpendicular to the base of the square:

$$(AC)=1=\sqrt{(CB)^2+(AB)^2} \text{ and } (CB)=(AB)$$

Therefore

$$(AB)=\sqrt{\frac{1}{2}}=\frac{1}{\sqrt{2}}$$

The construction shown in *Fig.* 7 is used to construct the root 1/3 from the root 1/2, the proof of the construction being,

$$(AE):(AD)::(AB):(AC), \quad (AB)=\frac{1}{\sqrt{2}},$$

$$(AC)=\sqrt{1+\frac{1}{2}}=\frac{\sqrt{3}}{\sqrt{2}}, \ (AD)=1$$

Therefore

$$(AE)=\frac{1}{\sqrt{2}}\times\frac{\sqrt{2}}{\sqrt{3}}=\frac{1}{\sqrt{3}}$$

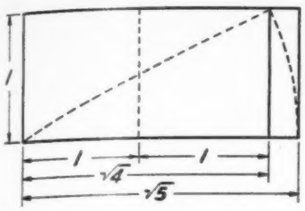


Fig. 5—Root 5 rectangle has diagonal of root 4 rectangle for side

The remaining reciprocal rectangles may be constructed similarly, *Fig.* 8 showing the construction of the root 1/5 rectangle from the root 1/4. Since $1/\sqrt{4} = \frac{1}{2}$, the

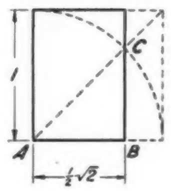


Fig. 6—Reciprocal root 2 rectangle is developed as shown

root 1/4 can be readilv constructed by dividing a square into two equal rectangles.

At first thought, the reciprocal-root rectangles offer no advance in design technique over the simple-root rectangles, since they are similar in shape. However, this is the one special similar rectangle that is used in the division of rectangles and in the application of dynamic symmetry to three-dimensional design. Many designs of ultimate complexity may be produced by the division of rectangles and reciprocals or the combination of rectangles and reciprocals.

A complement rectangle is that which added to the original will again form a square. In this case, the term complement implies a balancing of the areas. A reference to *Figs.* 6, 7, and 8, will show the relation of these complement rectangles to the original reciprocal-root rectangles, the length of the complementary sides being $1-1/\sqrt{2}$, $1-1/\sqrt{3}$ and $1-1/\sqrt{5}$ for the complements of the root 1/2, root 1/3, and root 1/5 rectangles, respectively. The complementary sides of the root 2, root 3, and root 5 rectangles will in like manner be equal to $\sqrt{2}-1$, $\sqrt{3}-1$, and $\sqrt{5}-1$ respectively. From the similarity of the figures, a general statement might be made that the complement root N is similar to the complement root $1/N$, but it should be noted that the complement of a reciprocal is not the reciprocal of a complement. The complement is also important in the formation of complex rectangles.

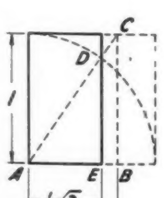


Fig. 7—Reciprocal root 3 rectangle

Basic rectangles consist of the square, the root rectangles, the reciprocal root rectangles and the complement rectangles. These form a foundation for the development of dynamic symmetry as a basis of design and, in themselves, offer a rich field for the machine designer to build on.

The late Jay Hambidge discovered and used dynamic symmetry to do research work in the analysis of Greek and Egyptian art, and the resulting findings bring back to us knowledge of design that had been forgotten and lost. In the work of analysis, Mr. Hambidge encountered a far more difficult problem than that of design. He

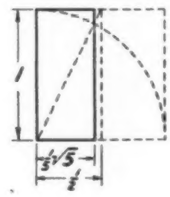


Fig. 8—Reciprocal root 5 rectangle developed from half-square, the latter being a reciprocal root 4 as well as being an arithmetic proportion

therefore resorted to arithmetical terms to assist him in making and comparing measurements. These numbers tend to confuse the understanding of the theory by mixing arithmetic into a study more easily grasped by the understanding of geometry. However, the values are shown

Dimensions of Basic Rectangles*

(Based on unit square)

Root	Simple-Root-Rectangles		Reciprocal-Root Rectangles	
No.	Basic	Complement*	Basic	Complement
Square	1.000	0	1.000	0
2	1.414	.414	.707	.293
3	1.732	.732	.577	.423
4	2.000	1.000	.500	.500
5	2.236	1.236	.447	.553

*Dimension of second side is unity except for the complement of the simple-root rectangle. This complement has for this dimension the simple-root rectangle side as tabulated.

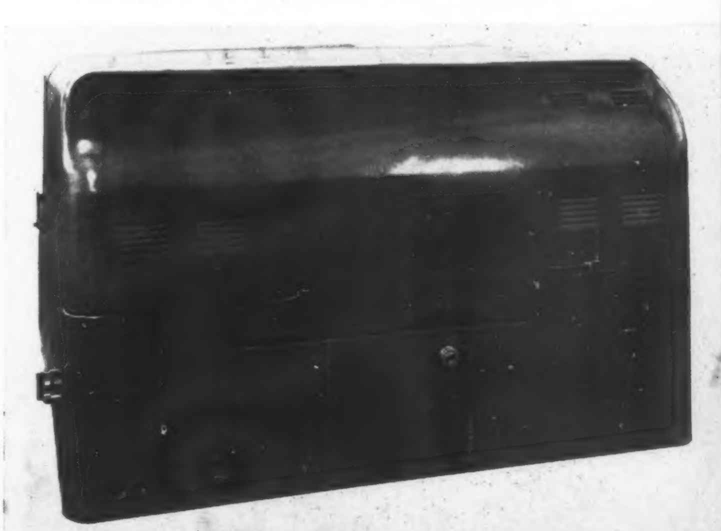
in the above tabulation; they were useful in analysis, they are used by many artists, so they may be of value to machine designers.

Design of the Landis grinding machine, *Figs.* 1 and 9 is based on the root 3 rectangle shown in *Fig.* 4. Various parts of the machine also use the root 3 relationship, the shape of the machine being repeated in the bed door and headstock as shown in *Fig.* 1.

The appearance of this large and complex machine tool has been determined mainly by the functional design. It illustrates how styling by means of dynamic symmetry provides unity to the design without conflicting in any way with these functional requirements.

(*Continued in next issue*)

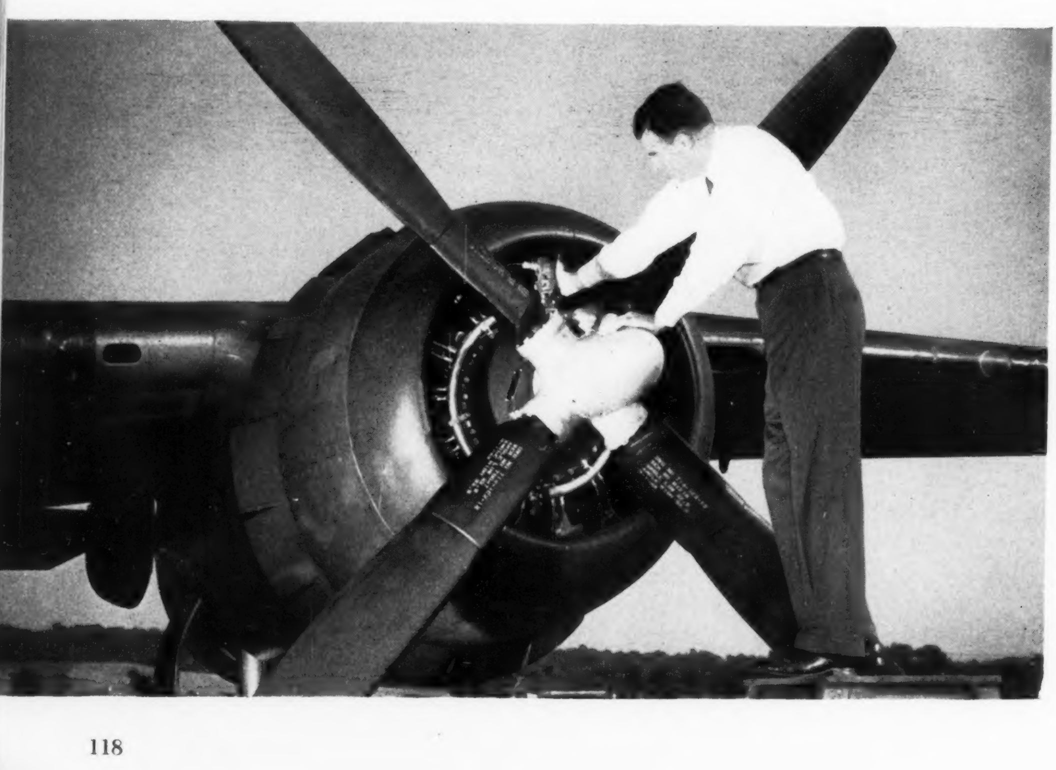
Fig. 9—Rear view of grinder shown in Fig. 1 illustrates the root 3 rectangle in its basic shape



Scanning the field for IDEAS

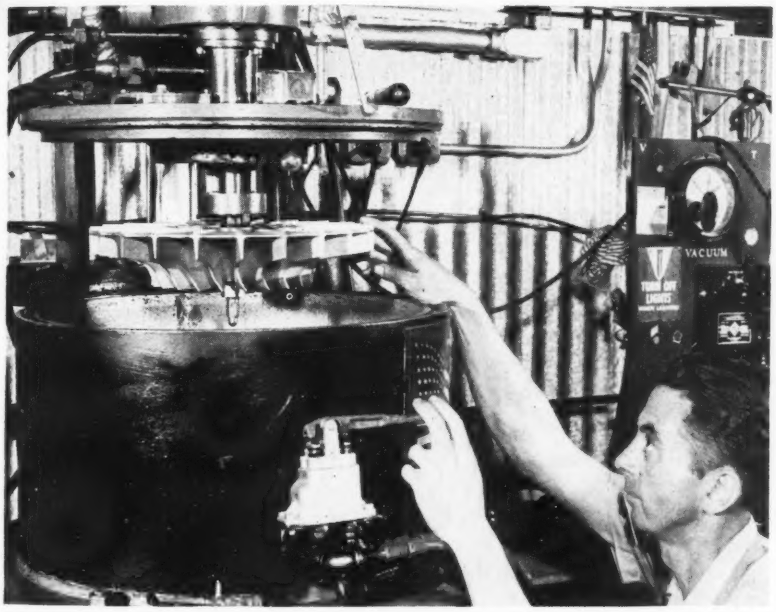
WIRE brushing wheel, left, has relieved a bad bottleneck in construction of General Sherman tanks. These wheels have eliminated a costly hand operation in the construction of tractor treads by cleaning out the rubber that flows into the end holes when the rubber is vulcanized to the metal. Strong adherence of the rubber complicated the hand operation. The Osborn Manufacturing Co. devised the wire wheel with a diameter equal to the inside diameter of the end holes so that, with the brush mounted on a grinding head, the holes can be quickly cleaned in a second or less. Containing .014-inch diameter steel wire, the brush is coarse enough to remove the rubber but not hard enough to mar the metal.

Engine synchronization for multiengine warplanes is facilitated by the development of a small alternator for mounting on each engine, all of which are pilot-controlled from a single knob. George W. Brady, chief engineer of the Curtiss-Wright propeller division, is shown at left inspecting the installation of one of these units. By utilizing the alternating-current principle, mechanical connections generally required with individual constant-speed governors have been eliminated. Also, being far simpler, the unit compares favorably from weight and cost standpoints.



Corrugated belt conveyors are useful in processing machines where it is desired to provide positive feed, equal spacing or separation of articles being manufactured. In the electrode coating machine, designed by

the Moslo Machinery Co., the rod conveyor, right, transfers coated electrodes from an extrusion coating unit to a brushing machine which removes one inch of the coated flux from one end of the rod. To assure positioning endwise, two converging end belts center the rods on the conveyor so that the brushing operation is uniform. The machine illustrated is designed to produce 350 rods a minute.



High vacuum is utilized in the test unit, above, to reduce the power required to drive supercharger impellers at 20,000 revolutions per minute. The terrific centrifugal stresses caused by spinning so fast make the aluminum alloy impellers take on a permanent set and become .003-inch larger in diameter. Stress-coat patterns showing regions and extent of stresses have been obtained on these General Electric impellers with the surface properly coated.

Simplification in design has effected marked economies in the Gorton etching machine, right. Earlier models utilized a complicated casting for the machine head which could be used both for the electric etcher and the engraving machine by assembling different units on the head. Standardizing in this way saved the costs of extra patterns and held inventory stocks to a minimum, as well as permitted users to convert one machine to another readily. A study of redesign possibilities, however, indicated the desirability for simplification of the etcher unit —even at the expense of standardization — resulting in reduced cost and material saving. The new machine, being a single-purpose model, is neater in appearance through the elimination of mountings not needed. These include omission of motor bracket and engraving-head ears.

Ceramic Parts in Design

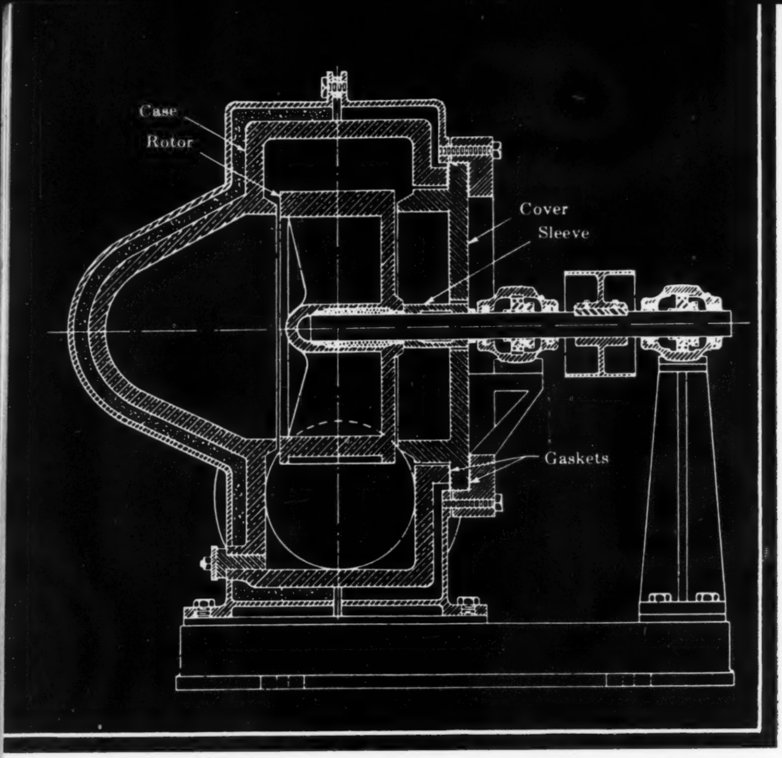


Fig. 1—Exhauster fan for handling corrosive gases has rotor, case, cover, and sleeve of de-aired chemical stoneware

AMONG the few engineering materials listed by the War Production Board as available in quantities significant enough to enable them to serve as substitutes are the ceramics. With densities somewhat less than that of aluminum, and specific strengths (strength divided by specific gravity) comparing favorably with cast iron, engineering grades of ceramic materials resist abrasion, corrosion, heat, moisture and weathering better than most metals. In addition, they possess high electrical resistance. In the chemical and electrical applications for which their characteristics qualify ceramics, mechanical strength is generally an essential requirement. There is therefore available to the designer a range of specially developed materials with proved mechanical properties which can successfully replace metals, hard rubber and plastics for many machine applications where shock hazards are not an important factor. Because of their exceptional resistance to chemical action of almost any kind they can replace stainless steels and other acid-proof alloys.

It is the purpose of this article to assist designers in developing and specifying ceramic parts for machines. Preceding a discussion of the factors which must be considered in the design of such parts, the materials available and methods of production will be briefly outlined.

Ceramic parts are formed while in a plastic state by a variety of methods which include casting, compression molding, injection molding, extrusion and machining. The shaped parts are then fired at high temperature after which they may be accurately ground if necessary. By choosing the proper raw materials a wide range of bodies can be produced. While all possess in marked degree the properties of chemical resistance, electrical insulation and resistance to high temperature, actual values can be controlled within wide limits. In addition, parts may be made either porous or impervious to moisture or may be made resistant to thermal shock.

For convenience, ceramic materials suitable for mechanical requirements may be roughly classified as stoneware, porcelain and steatite. In general, these materials need not be regarded as competitive with each other because even for purposes where two or more may give equally satisfactory service, the cost of manufacture usually determines the selection. Stoneware has been developed primarily for chemical equipment (*Fig.* 1) while the porcelains and steatites find their principal application as electrical insulators (*Fig.* 2). Typical mechanical properties are listed in TABLE I.

Ceramic materials in general contain three types of constituents, each having a definite function in developing the desired properties in the finished part. These may be classified as: (a) Plastic constituents such as clay, (b) nonplastic or filler materials such as silica (flint or quartz), and (c) fluxes or solvents such as feldspar. At the high temperature of firing, which may reach 2300 degrees Fahr., thermal dissociation of the clay minerals forms crystalline needles of mullite and releases silica which reacts with feldspar to form a glass matrix (vitrification) in which the crystalline mullite and infusible silica from the initial mixture are embedded. Control of materials and extent of vitrification determine the properties of the finished part. For example, white-firing raw materials used in porcelain have somewhat lower dry strengths than the clays used in stoneware, which limits size of pieces that can be made.

How Parts May Be Shaped

After refining, the materials are mixed with water in the proper proportion, aged and treated to remove included air from the body, increasing toughness. In shaping the part either a wet or dry process may be used. The term is relative inasmuch as a certain minimum amount of water is necessary to give the clay body sufficient plasticity for forming, as well as strength for self-support prior to and during firing.

In the wet process, as applied to porcelain, the various ingredients are mixed with a sufficient quantity of water to result in a fluid mass. In this fluid mass the solid particles are completely dispersed so that by agitation it is possible to reach complete uniformity. Some of the particles when so dispersed are of colloidal size and the resultant liquid

Table I

Properties of Typical Ceramic Materials

Material	Stoneware (Chemical)	Porcelain (Wet Process)	Porcelain (Dry Process)	Porcelain (Prestite)	Steatite (AlSiMag 196)	Mica-Glass (Mycalex)
Tensile Strength* (psi)	7,500	7,000	3,000	5,000	10,000	5,000- 7,000
Compressive Strength* (psi)	117,000	50,000	30,000	48,000	85,000	22,000-30,000
Modulus of Rupture* (psi)	13,000	12,000	5,000	11,000	22,000	15,000-20,000
Specific Gravity	2.3	2.33	2.28	2.4	2.7	3.44
Coefficient of Thermal Expansion (Per °F)	.08 to 2.4 x 10^6	3.17 x 10^6	3.17 x 10^6	3.17 x 10^6	3.5 to 4.33 x 10^6	4.45 x 10^6

*Figures are comparative only, based on controlled tests.

mixture, known as "slip", partakes of some of the properties of a colloidal suspension. From this liquid slip, solid parts may be shaped by one of two basic methods.

The liquid slip may be cast directly into plaster of paris molds. In this case the moisture is absorbed from the slip by the porous plaster of paris, leaving a solidified plastic form. After casting, the plastic shape must be removed from the mold and dried. In the drying operation, as the water is driven off there is a volumetric shrinkage in the part. If drying is too rapid or nonuniform the differential shrinkage will result in warpage or cracks. After drying, the piece may be machined if required, glazed, and placed in the kiln for firing. In the process of firing an additional shrinkage occurs. The glaze is fluxed down to produce a uniform glassy surface of the color desired, and the body is vitrified. Wet process cast porcelain such as this is nonporous, dense and uniform.

Excess Moisture Is Removed

The liquid slip may also be reduced to a solid state by what may be called plastic working. In this operation the slip is pumped into filter presses in which the excess moisture is removed, leaving the clay in a plastic or putty-like state. This material is then run through vacuum pug mills in which it is shredded, de-aired and extruded as a tight form, the shape depending on the nozzle of the pug mill. For solid, round or rectangular pieces, tubes, etc., the product of the pug mill may be used directly without further working. In other cases it may be necessary to take the pugged clay and press it into shape in plaster of paris molds (plunging), or turn it to shape by any one of a variety of methods. The plastic part so formed must then be dried and after drying can be further machined, glazed, etc., and fired in the same manner as cast porcelain. Plastic porcelain made by the above process also is dense, nonporous, and uniform in structure.

Porcelain may also be prepared by adding to the basic constituents only a relatively small amount of water, not sufficient to result in a fluid mass. These components are then thoroughly agitated together in a dry mixer. In this process it is not possible to get as uniform a dispersion of the various phases as with the wet process. The average particle size in the aggregate tends to be larger, consequently the dry process body tends to be coarser and less uniform in structure. After pressing it may be machined if desired, glazed and fired. Dry process porcelain made by this method may be slightly more porous and both its mechanical and electrical strengths are inferior to those of wet process porcelain.

Great improvements may be made in the general quality of dry process porcelain by the use of vacuum dies, high pressures and other refinements. A group of parts made by such methods is shown in *Fig.* 2. The term dry process porcelain may therefore cover a multitude of different qualities of material. In general the dry press parts are cheaper than wet process, while dimensional tolerances may be held closer.

Stoneware parts are made by machining (plastic or dry) by molding and by "turning" (plastic forming on a potter's wheel). By plastic pressing, utilizing mainly hand work, large pieces may be fabricated. Following the shaping of the part, time must be allowed for drying, large pieces requiring a month or more at this stage. Shrinkage amounting to as much as 1 inch per foot occurs, making close size and shape tolerances difficult to maintain.

Steatite parts usually are made from a synthetic blend of specially prepared constituents including clay, talc and fluxing materials. Parts may be produced on high-speed presses or extruded, also may be machined if necessary prior to firing. Natural steatite (soapstone), which is a massive form of talc, may be machined while in its soft natural state, then kiln-fired to a high degree of hardness after which further machining is impossible. In such cases the shrinkage during firing is relatively small.

Where close dimensional accuracy is required, surfaces of fired ceramic parts may be finished by grinding to

Fig. 2—Pressure-molded porcelain parts may be produced in intricate shapes for mechanical and electrical purposes

tolerances as close as 1/10000-inch. By this means, valves such as the one shown in *Fig.* 3 may be made fluid-tight against pressure as high as 60 pounds per square inch or more. Unground porcelain parts produced by the wet process can be held to a differential tolerance of plus or minus 1/32-inch per inch with a minimum of plus or minus 1/32-inch for smaller sizes. Since the shrinkage of dry pressed porcelain is less than that of wet process porcelain, it may be held to closer dimensional tolerances. In this case the tolerance is of the order of 1/100-inch per inch.

Lack of Ductility Important Property

In the mechanical design of ceramic parts one of the most important factors for the designer to bear in mind is the lack of ductility of the material. In dealing with most metals a slight overstress in the material at some particular fiber location results merely in the elongation of that fiber and consequent redistribution of the load over the section. Ceramics in general do not possess the ability to yield and redistribute the load, and for virtually all such materials the elastic limit and ultimate strength are the same. As soon as any fiber anywhere in the body is stressed beyond its elastic limit a fracture or crack occurs at that point. If the load persists, the actual unit stress in the fibers at the base of the crack may become theoretically infinite so that the crack progresses through the piece until ultimate fracture results. For this reason the method of applying the load to any ceramic part is of great importance. Concentrated loads applied at any one point are in all cases undesirable.

So far as is known, ceramics are not subject to fatigue as normally understood in metallurgy. The subject has not been thoroughly and completely investigated, but within the limits of practical experience there has never been any indication of fatigue failures in such materials. Also, there is no evidence of any change in the state or mechanical properties with age.

Surfaces of ceramic materials may be either glazed or unglazed. The purpose of glaze is usually to improve appearance and make surfaces easier to clean. It should be noted that inasmuch as most ceramic materials are nonporous, glaze is not necessary to insure resistance to penetration of moisture. Two types of glaze may be used. Slip glaze is liquid glaze which is applied to the body before firing or to ware that has been fired once (biscuit fire) and is then fired again (glaze fire). Salt glaze used on stoneware is achieved by throwing salt on the fire at the end of the firing cycle. If the glaze is properly fitted to the body it actually increases strength.

To understand the effect of glaze on strength it should be remembered that ceramic bodies are strong in compression but relatively weak in tension. Consequently under mechanical loading virtually all failures are of a tensile nature. Even though a short cylinder is loaded in direct compression, the ultimate fracture is usually a tension break occasioned by the hoop tension set up in the piece resulting from a compressive deformation. Under most loading conditions the maximum fiber stress is developed in one of the surfaces. If an initial compressive force can be set up in this surface, then before fracture occurs the resultant unit fiber stress in that surface must be sufficient to overcome the initial compression and reach the tensile elastic limit of the fiber.

This is actually accomplished by applying to the piece a glaze which has a lower coefficient of expansion than the body itself. Consequently in cooling from the 2200-degree firing temperature the glaze tends to shrink less than does the underlying body. However, since the glaze is integrally bonded to the body there can be no movement between the two and the glaze is actually placed in compression by the shrinkage of the body. The degree of increase in strength achieved by the selection of the proper glaze varies over a wide range depending upon the shape of the body and the nature of loading. In some cases the increase in strength achieved by proper glazing may be more than 100 per cent. With an improper glaze, having a higher coefficient of expansion than the body, the mechanical strength of the finished piece may be actually reduced below that of an unglazed body.

Most of the highly vitrified ceramic materials such as porcelain may be readily broken by sudden and extreme temperature change. The thermal shock resistance of porcelain is substantially better than that of ordinary glass and is of about the same order as that of borosilicate glass. In general, almost any reasonable porcelain shape will withstand, without failure, immediate immersion from boiling water to ice water, or vice versa. However,

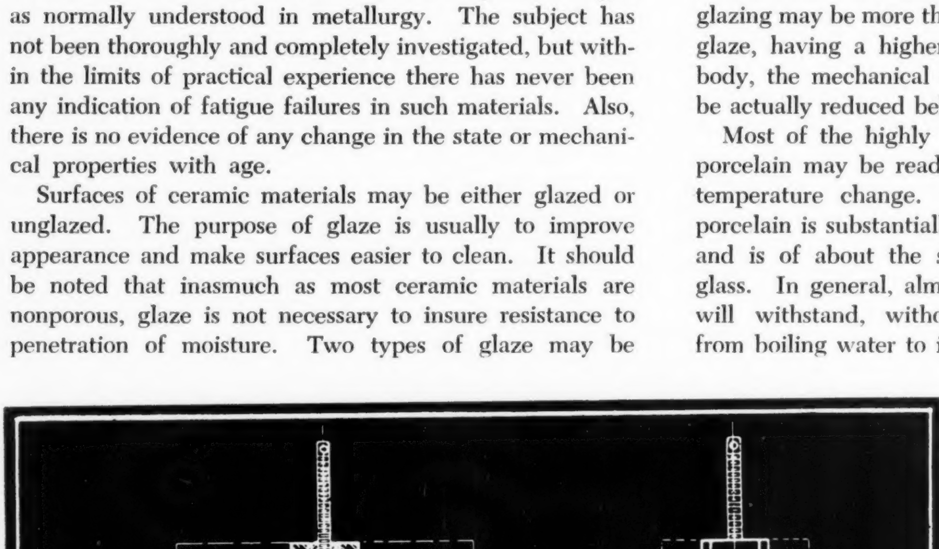


Fig. 3—Stoneware plug and body of lubricated valve are ground to an accurate fit

as the temperature differential increases above the 180-degree spread between boiling water and ice water, the possibility of fracture due to thermal change increases. Porcelain is capable of operating at fairly high temperatures without mechanical failure provided it is not subjected to sudden temperature change while under working conditions. The exact limitation on thermal shock which can be withstood by any particular piece is, of course, a function of the shape and design of the piece. When porcelain is assembled with hardware or other materials having a different coefficient of expansion, the shock resistance of the assembled piece may be materially reduced due to the differential expansion or contraction between the various elements.

In the case of regular stoneware, actual working temperatures should not exceed 200 degrees Fahr. In this type of material the three phases (mullite, quartz and silicate glass) have widely different thermal expansion coefficients, and repeated heating and cooling weakens the bond between the glass and crystalline substances. Newly developed heat-shock resistant bodies provide pore spaces into which single constituents can expand, decreasing the effect of differential thermal expansion but increasing the water absorption. Some of these materials have working temperatures far in excess of 200 degrees and may be plunged while white hot into ice water without experiencing damage.

Thermal conductivity of heat-shock resisting stoneware used for hot liquids and gases is less than one-tenth that of metals, resulting in low heat loss from pipelines of this material. However, heat transfer through walls of heat exchangers is slow and for such applications special stonewares of high heat conductivity have been developed.

Design Factors To Be Considered

In the actual design of ceramic parts the peculiarity of the material and limitations imposed by the method of manufacture must be carefully considered. Before any design is completed it is highly desirable that the ceramic manufacturer be consulted. His suggestions may result in both improved strength and saving in cost. In working up the preliminary design, the engineer should pay particular heed to the following general considerations:

Safety factors should be higher than for metals because of the impossibility of maintaining uniformity. Also, average strength tests may vary as much as 10 per cent due to localized inherent weakness in individual pieces. Because ceramic materials lack ductility, the influence of stress concentration must weigh heavily in selecting safety factors. Sudden impact loads should be avoided.

To develop maximum strength under any particular set of conditions, concentrated loads must be obviated. High unit loading stress due to uneven or rough surfaces lowers overall strength and can be avoided by grinding the load-bearing surfaces and by gasketing.

Wall thickness should be as uniform as possible, otherwise internal strains will be set up which cause cracks.

Corners should be filleted or chamfered to avoid stress concentration and to obviate damage to the unfired piece.

Possible warpage of the piece must be allowed for, particularly on wet process parts. In general, the more liberal the tolerances the lower will be the cost. Glazing reduces the dimensional accuracy of the part.

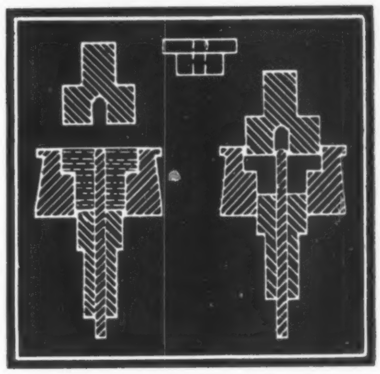


Fig. 4—Dies for pressing shoulder bushing of steatite employ a stationary pin to produce the central hole

Draft should be allowed on parts which are pressed, so that they may be withdrawn from the mold or die.

Parts may be attached to each other in various ways. In the case of stoneware, parts should be cemented together at specially prepared joints, avoiding the use of bolt holes. Porcelain, however, may be drilled and tapped with considerable accuracy. Screws of brass or other soft material are preferable to steel.

Bonding other parts to porcelain is possible provided the surfaces are rough. The commonest and most satisfactory method of obtaining this roughened surface is by applying what are called sand bands. In performing this operation, sand-like particles of ceramic material are bonded to the desired surface with clay. In design, allowance should be made for about 1/16-inch thickness of such a sand belt. The hardware or other parts may be attached to these sand bands by portland cement or type metal. With such assemblies care must be exercised to provide for the differential thermal expansion between the porcelain and the attached parts. Also, the design of the joint should be carefully made to distribute the mechanical stress uniformly across the joint.

Porcelain parts can be produced with metallized and tinned surfaces with which it is possible to make tight solder joints. It is also possible to furnish porcelain glazed with high-resistance conducting material. Also, two or more porcelain elements can be bonded together integrally to form one solid ceramic piece by means of glazed joints. Such joints must be made in the piece at the time of the initial firing.

In conclusion, it should be pointed out that the ceramic manufacturer is in a much better position to furnish satisfactory parts if the designer supplies complete information regarding use for which the part is intended, critical dimensions, amount and nature of loading, temperature of operation, atmospheric conditions, wear conditions, size and shape of adjacent metal or other part, on what surfaces glaze is desired, etc. Many ceramic bodies have been developed to meet special requirements and recommendation of the particular materials best suited to a specific application should be left to the manufacturer.

MACHINE DESIGN gratefully acknowledges the cooperation of the following companies in the preparation of this article: The Akron Porcelain Co.; American Lava Corp. (*Fig.* 4); The Colonial Insulator Co.; Locke Insulator Corp.; General Ceramics Co.; General Electric Co.; The Star Porcelain Co.; Stupakoff Ceramic & Mfg. Co.; The United States Stoneware Co. (*Figs.* 1 and 2); and Westinghouse Electric & Mfg. Co. (*Fig.* 2). The generous assistance of the Locke Insulator Corp. and the United States Stoneware Co. is particularly appreciated.

Designing RING SPRINGS

By A. M. Wahl
Westinghouse Research Laboratories

—Photo, courtesy United Engineering and Foundry Co.

Fig. 1 Left—Ring-spring nest used in the rotating block shown in Fig. 2. Weighing 1500 pounds, this spring has an ultimate resistance of 320,000 pounds

Fig. 2 Above—Block for a forging press utilizes a ring-spring nest, comprising two complete springs working one within the other

WHERE space is limited and a relatively large amount of energy must be absorbed, a type of spring known as the "ring spring" may well merit consideration by the designer (1)*. This is particularly true if the application is one where a large amount of damping is also desirable, such as for railway draft gear springs (2).

In *Figs.* 1 and 2 is shown an application of a ring spring to a rotating block. The purpose of this device is to rotate heavy forgings by actuating the chain which supports one end of the forging while the other is being worked under a hammer or press. This block utilizes a ring-spring nest. One spring is shown in *Fig.* 1, the other being tandem-connected with and "telescoped" in the spring shown. This is probably the highest capacity spring ever made, the ultimate resistance of the spring being 320,000 pounds with a total travel of 20 inches. Weight of spring alone is 1500 pounds.

As its name implies the ring spring consists essentially of a series of rings having conical surfaces and assembled as indicated in *Figs.* 3 and 4. When an axial load is applied, sliding occurs along the conical surfaces with the result that the inner rings are compressed and the outer rings extended. In this manner an approximately uniform distribution of circumferential stress is obtained in both the inner and outer rings. Because of this uniformity of stress distribution, the ring spring is commonly assumed to act essentially as a bar in simple tension and to have a correspondingly high efficiency (considered on the basis of allowable energy storage per pound of metal). Actually, because of compression stresses at the conical surfaces of the outside rings, there will be a slight nonuniformity in *equivalent stress* distribution. Where a tension and compression stress act at right angles as in this case, the *equivalent stress*—on the basis of the maximum shear theory of strength—will be the sum of the numerical magnitudes of these tension and compression stresses.

In addition, where the radial thickness of the rings is appreciable, there is some nonuniformity in circumferential stress since the ring behaves like a thick cylinder under internal or external pressure. For most springs, however, this nonuniformity in equivalent stress distribution will not be large and hence this type of spring will have a relatively high efficiency. On the other hand, it should be noted that the damping in this spring is obtained at the expense of a certain amount of wear on the sliding surfaces, even though lubricated in accordance with usual practice.

A typical load-deflection hysteresis loop for the ring spring is shown in *Fig.* 5. From this it may be seen that on the compression stroke a much higher spring constant (in terms of pounds per inch deflection) is obtained than for the return stroke. This is due to the friction forces on the conical faces of the rings which, for an increasing load, are added to the elastic forces caused by distortion of the rings but for a decreasing load are subtracted from the elastic forces. Thus a large hysteresis loop is obtained

*Numerals in parentheses refer to list of references at end of article.

with correspondingly high energy absorption per cycle.

Referring to *Fig.* 6, for practical purposes of analysis each conical surface of the ring spring may be considered as subject to a total normal force N distributed uniformly around the circumference and a friction force $F = \mu N$ (when μ is the coefficient of friction). This latter force acts in the direction shown when the spring is being compressed, and in an opposite direction when the spring is being extended. These forces N and F produce primarily a compression of the ring, although there is at the same time a tendency of the ring to bend like a bar on elastic foundation (3). This latter effect, however, may be neglected for practical design purposes. It will also be assumed that the ring thickness is small compared to the mean diameter so that the nonuniform circumferential distribution due to the thick cylinder effect may be neglected (4).

Considering the inner ring of *Fig.* 6 and assuming the spring is being compressed so that the friction force acts

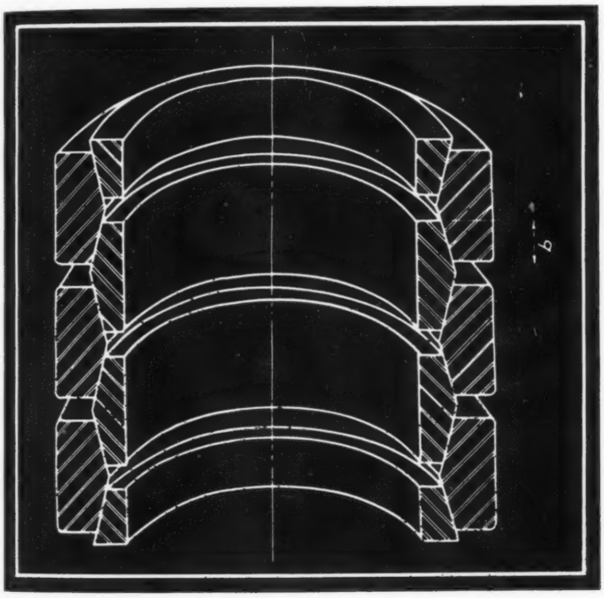


Fig. 3—Diametral section through ring spring

in the direction shown, the total radial force acting will be equal to $2(N \cos\alpha - F \sin\alpha)$ where α is the angle of taper of the conical surfaces. The radial load p per inch of the circumferential center line of the ring will be the total radial force divided by $2\pi r_i$, where r_i is the mean radius of the ring. Hence

$$p=\frac{2(N\cos\alpha-F\sin\alpha)}{2\pi r_i}$$

Taking $F = \mu N$ this equation becomes

$$p=\frac{N(\cos\alpha-\mu\sin\alpha)}{\pi r_i} \qquad (1)$$

For a thin ring, the compressive stress will be

$$\sigma_c=\frac{p r_i}{A_i} \qquad (2)$$

where A_1 = sectional area of the inner ring. Substituting Equation 1 in 2,

$$\sigma_c=\frac{N(\cos\alpha-\mu\sin\alpha)}{\pi A_i} \qquad (3)$$

The axial load P acting on the spring during the compression stroke is found by taking the components of N and F along the axis, *Fig.* 6. Hence

$$P=N\sin\alpha+F\cos\alpha=N(\sin\alpha+\mu\cos\alpha)$$

Solving this for N and substituting in Equation 3, the circumferential compressive stress σ_c in the inner rings becomes

$$\sigma_c=\frac{P}{\pi A_i}\,\frac{\cos\alpha-\mu\sin\alpha}{\sin\alpha+\mu\cos\alpha} \qquad (4)$$

This equation may be reduced to the simpler form:

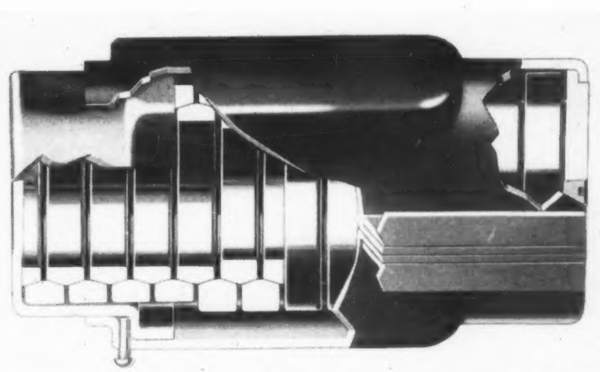
$$\sigma_c=\frac{P\tan\alpha}{\pi A_i K} \qquad (5)$$

where

$$K=\frac{\tan\alpha(\mu+\tan\alpha)}{1-\mu\tan\alpha} \qquad (6)$$

To facilitate practical use of Equation 5, values of K are plotted against the angle α for various friction coefficients μ in the upper group of curves of *Fig.* 7. Where maximum accuracy is desired, computation should be made by using Equation 6.

A similar procedure for calculating the circumferen-



—Photo, courtesy Edgewater Steel Co.

Fig. 4—Sectional view of ring spring as used in draft gear where a large amount of energy must be absorbed in a relatively small space

tial tension stress σ_t in the outer rings is used. This gives

$$\sigma_t=\frac{P\tan\alpha}{\pi A_o K} \qquad (7)$$

where in this case A_o = sectional area of outer ring and K is given by *Fig.* 7. It should be noted that Equation 5 and 7 give the stresses in the spring as a function of load for increasing load P only.

As indicated previously, to obtain the *equivalent stress* in the outer rings, the compressive stresses due to the

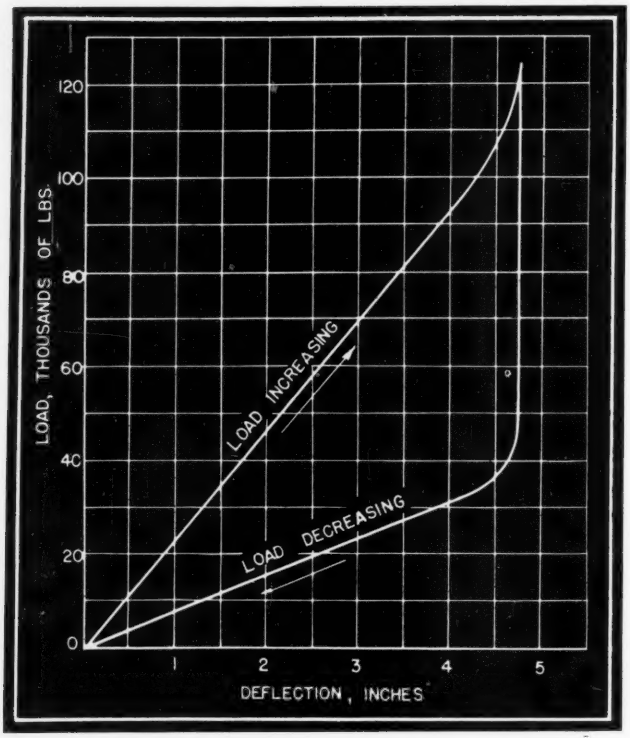


Fig. 5—Typical load-deflection diagram of ring spring showing hysteresis loop

normal forces N, *Fig.* 6, should be added to the circumferential tension stress σ_t calculated from Equation 7. These compressive stresses may be computed as follows: For the outer ring the radial load p per inch of mean circumference is obtained from Equation 1 using r_o instead of r_i. Thus

$$p=\frac{N(\cos\alpha-\mu\sin\alpha)}{\pi r_o}$$

Solving for N, this gives

$$N=\frac{p\pi r_o}{\cos\alpha-\mu\sin\alpha}\quad\text{(8)}$$

If σ_t is the tension stress, from the usual ring formula:

$$p=\frac{\sigma_t A_o}{r_o}$$

Substituting in Equation 8,

$$N=\frac{\sigma_t A_o \pi}{\cos\alpha-\mu\sin\alpha}\quad\text{(9)}$$

Letting b equal the projected axial length of contact area at the load P and stress σ_t, *Fig.* 3 (which length may be obtained from the calculated deflection of the spring and its geometrical proportions), then the average compression stress σ_c' in the contact region is

$$\sigma_c'=\frac{N\cos\alpha}{2\pi r_m b}\quad\text{(10)}$$

where $r_m = (r_o+r_i)/2$ = mean radius of inner and outer rings. This holds since the total area over which the force N acts is $2\pi r_m b/\cos\alpha$. Substituting in this the value of N given by Equation 9 the contact stress becomes

$$\sigma_c'=\frac{\sigma_t A_o}{2r_m b(1-\mu\tan\alpha)}\quad\text{(11)}$$

To calculate the total deflection of the spring, the radial deflections of the rings must first be found. For the inner ring the radial deflection will be approximately equal to $\sigma_c r_m/E$ where E is the modulus of elasticity. The axial deflection of the spring due to each inner ring will be two times the radial value $\sigma_c r_m/E$ divided by $\tan\alpha$. (The factor two is used since there are two conical surfaces per ring). Hence if n_i is the total number of inner rings in the spring (a ring of half the full section being considered as half a ring), the deflection δ_i due to these rings only will be

$$\delta_i=\frac{2\sigma_c n_i r_m}{E\tan\alpha}\quad\text{(12)}$$

Similarly the total axial deflection δ_o due to the outer rings only is

$$\delta_o=\frac{2\sigma_t n_o r_m}{E\tan\alpha}\quad\text{(13)}$$

where n_o is a total number of outer rings. This will also be equal to n_i.

Letting $n=n_o+n_i=$ total number of "elements" in the spring, each element consisting of a half inner and a half outer ring, and adding Equations 12 and 13,

$$\delta=\delta_o+\delta_i=\frac{r_m n}{E\tan\alpha}(\sigma_t+\sigma_c)\quad\text{(14)}$$

Using values of σ_c and σ_t given by Equations 5 and 7 in this, the deflection during the compression stroke may be expressed, in terms of the load P,

$$\delta=\frac{Pr_m n}{\pi E A_i K}\left(1+\frac{A_i}{A_o}\right)\quad\text{(15)}$$

In this K is given by Equation 6 or *Fig.* 7.

The deflection and loads occurring during the unloading or return stroke may be analyzed in a similar manner by considering that in this case the direction of the friction forces F, *Fig.* 6 will be reversed. If P_1 is the load

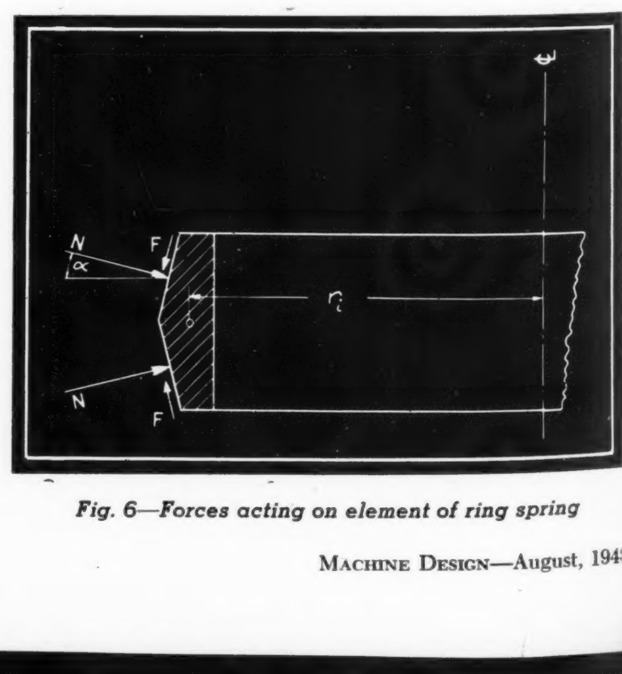


Fig. 6—Forces acting on element of ring spring

and δ_1 the corresponding deflection during the return stroke,

$$\delta_1=\frac{P_1 r_m n}{\pi E A_i K_1}\left(1+\frac{A_i}{A_o}\right) \quad \text{(16)}$$

where

$$K_1=\frac{\tan\alpha(\tan\alpha-\mu)}{1+\mu\tan\alpha} \quad \text{(17)}$$

Values of K_1 are plotted for convenience in the lower group of curves of *Fig.* 7.

The ratio between the load P_1 (return stroke) and the load P (compression stroke) at any given deflection is

Load and Deflection Ratings

Inches				Pounds			
O.D.	I.D.	P_1	δ/n	W	P	P_0	P_1
3.750	3.020	.750	.250	.778	1135	750	382
3.093	2.240	.540	.152	.480	1420	915	450
1.521	1.275	.1750	.0265	.0286	3000	2300	1630
2.124	1.856	.230	.041	.0378	3250	2340	1465
.980	.724	.1785	.0164	.0179	3500	2440	1430
1.813	1.498	.204	.0366	.0522	5000	3400	1880
1.906	1.530	.2000	.0323	.0602	5000	3825	2720
4.305	3.982	.420	.085	.1436	5769	4150	2600
1.917	1.530	.2000	.0348	.0625	6900	4450	2970
2.242	1.785	.2250	.0378	.0961	6800	5200	3700
1.875	1.327	.200	.0344	.0896	7000	5660	4390
2.048	1.660	.2635	.0376	.0889	7900	5870	3910
2.587	2.040	.2500	.0434	.1469	9000	6900	4900
1.470	.880	.212	.0248	.0668	9420	6410	3540
3.152	2.550	.3000	.0537	.2409	12000	9200	6500
3.441	3.024	.360	.0719	.2367	12150	8250	4550
1.875	1.203	.290	.0316	.1365	13875	11350	9000
3.549	2.800	.325	.0595	.357	16000	12250	8700
3.441	2.419	.275	.0560	.375	22500	15670	9150
3.441	2.353	.375	.0575	.540	27750	22750	18000
4.375	2.976	.312	.0830	.732	32000	21500	11400
3.663	2.302	.500	.059	.916	50500	37400	25000
4.938	3.134	.375	.0830	1.235	52000	35800	20400
5.625	3.758	.645	.0800	2.312	97000	70000	44600
8.188	6.496	.786	.1680	4.57	100000	67200	35800
7.500	5.189	.692	.1215	4.61	110000	76000	43000
6.500	3.881	.593	.110	3.64	128000	85500	45000
8.188	4.599	.692	.1281	7.08	186000	128000	73000

*Ring Spring to be guided, either on the outside in a cylinder of 2% greater diameter than the O. D. or on the inside by a mandrel of 2% smaller than the I. D. of the spring.

O.D. = Outside diameter of spring.
I.D. = Inside diameter of spring.
B_1 = Solid height of one spring element (—one-half the width of one outer ring).
δ/n = Travel of one spring element.
$B_1+\delta/n$ = Free height of one spring element.
W = Weight of one spring element
P = Compression force of spring.
P_0 = Elastic force of spring.
P_1 = Recoil force of spring.

obtained by equating δ and δ_1, Equations 15 and 16. This gives

$$\frac{P_1}{P}=\frac{K_1}{K} \quad \text{(18)}$$

Hence to find the ratio of the spring constants for the return and compression strokes respectively it is only necessary to take the ratio K_1/K for the given values of μ and α. This is true since the spring constants are proportional to the respective loads at any given deflection.

Design Calculation

As an example of the application of these formulas in practical design, a ring spring of the following dimensions as tested by Wikander (1) may be considered: $A_o=A_i=.584$ in.^{2}, $r_i=4.42$ in., $r_o=4.74$ in., $r_m=4.58$ in. $\tan\alpha=.25$, $\alpha=14^\circ$(approx.), $E=29\text{x}10^6$psi, $n_o=n_i=9$, $n=18$. Tests on this spring indicated a coefficient of friction $\mu=.12$. From *Fig.* 7, for $\alpha=14^\circ$ and $\mu=.12$, by inter-

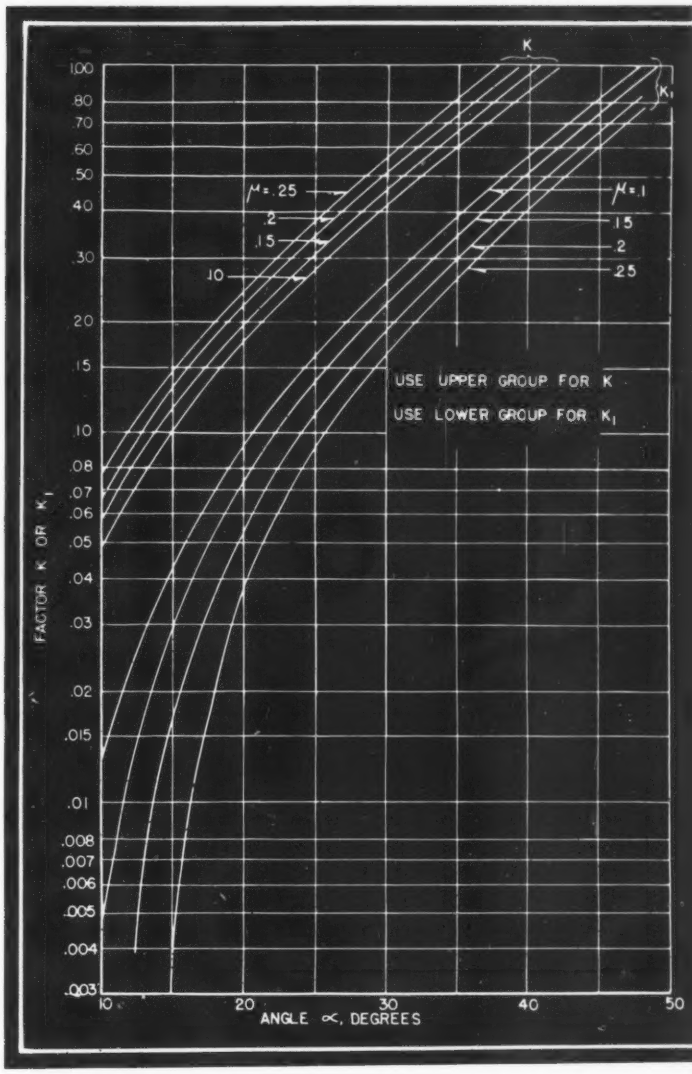


Fig. 7—Curves for finding factors K and K_1 in formulas

polation $K=.095$ and $K_1=.031$.

Assuming a maximum load $P_1=100{,}000$ pounds, then from Equation 15 for an increasing load the deflection δ is

$$\delta=\frac{100000\times4.58\times18\times2}{\pi\times29\times10^6\times.584\times.095}=3.26 \text{ in.}$$

The load P_1 on the return stroke for this same deflection will be equal to P multiplied by the ratio K_1/K, Equation 14.

This gives

$$P_1=100000\times\frac{.031}{.095}=32{,}700 \text{ lb.}$$

The spring constant for the compression stroke is

$$\frac{P}{\delta}=\frac{100000}{3.26}=30{,}700 \text{ lb/in.}$$

That for the return stroke is:

$$30700\times\frac{K_1}{K}=30700\times\frac{.031}{.095}=10{,}000 \text{ lb/in.}$$

(*Concluded on Page* 212)

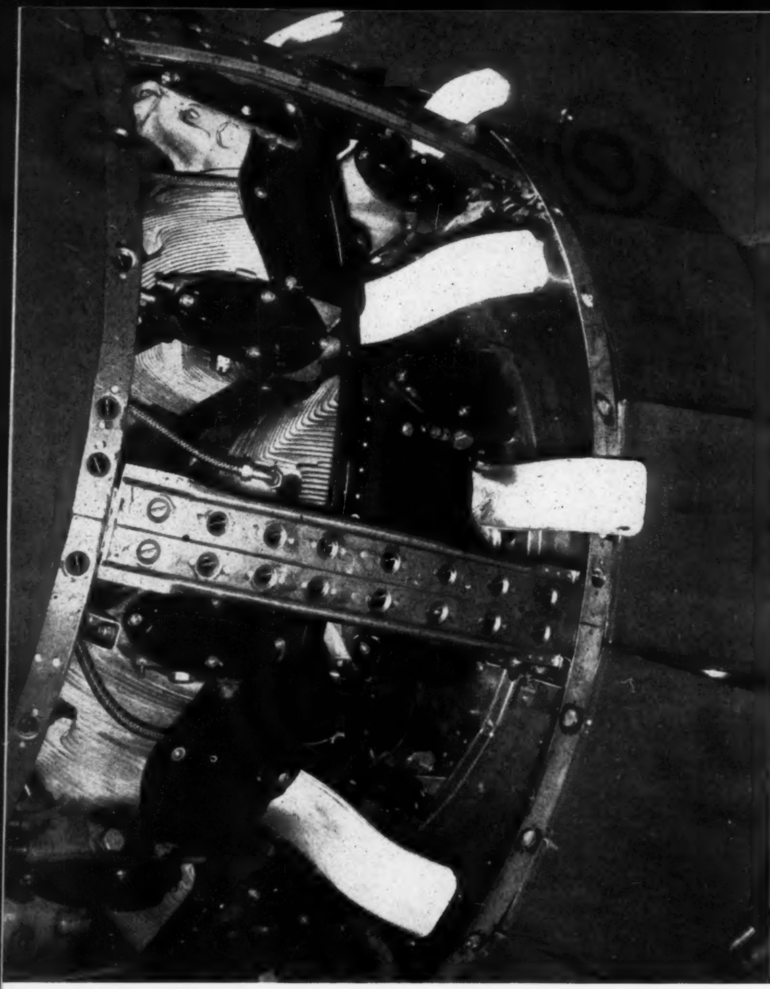


Fig. 1—Vitreous enameled SAE 1020 steel replaces stainless steel in the exhaust stacks for the famous B-25 Mitchell bomber

Substitution

By Irwin H. Such

INCREASED emphasis on the conservation and substitution of strategic materials makes it advisable for the design engineer not only to appraise the supply situation from time to time but to keep informed as to the trends in applications for materials in various types of equipment.

Conservation and substitution have become something more than a temporary war measure in that the lessons now being learned are exerting a permanent influence on design, first for the reason that many substitute materials previously thought inferior are performing nobly in service and, secondly, because in the postwar days to come there is every evidence that engineers will be under pressure to keep material and production costs at a minimum.

As an example, North American Aircraft Corp. has converted the entire exhaust assembly, *Fig.* 1, for the Mitchell B-25 bomber to vitreous enameled steel and Bell Aircraft Corp. is using some of the same material for short exhaust pipes for the Aircobra. Continental Motors Corp. has conducted tests in connection with its W-670 engine. Similar tests also are being conducted by Packard Motor Car Corp. and the Glenn L. Martin Co.

It will be recalled that Packard used enameled cast iron exhaust manifolds in its automobile engines with some success, the smoother surface providing freer flow of gases and added cleanliness. In applying enamel to steel aircraft exhausts, the problem largely involves obtaining closely adherent, thin coatings.

Unexpected changes sometimes occur in the material supply. Molybdenum, for instance, was pushed by the war agencies a number of months ago as a substitute for nickel and chromium in making alloy steels. More recently the situation has become reversed, the War Production Board ordering a reduction in production of molybdenum steels and an increase in the chromium-nickel types. An extremely critical situation in tungsten some months ago has been considerably relieved and by the same token tool steels now are in more adequate sup-

Fig. 2—Critical materials are saved in two ways in this gear, first by precision forging from a smaller blank and, second, by substituting an NE steel for a more highly alloyed steel

ply. Not only have sources for tungsten in the Americas been exploited but some metal even is arriving from China.

Despite added steelmaking capacity, Claimant Agencies such as the Army, Navy and Maritime Commission have entered combined requirements for the current (third) quarter exceeding production by roughly 1,000,000 tons and a similar deficit is seen for fourth quarter. Fortunately, the situation could be much worse as indicated by the fact that the steel industry has been able to make 275,-000 tons of National Emergency steels available on

Widen Design Horizons

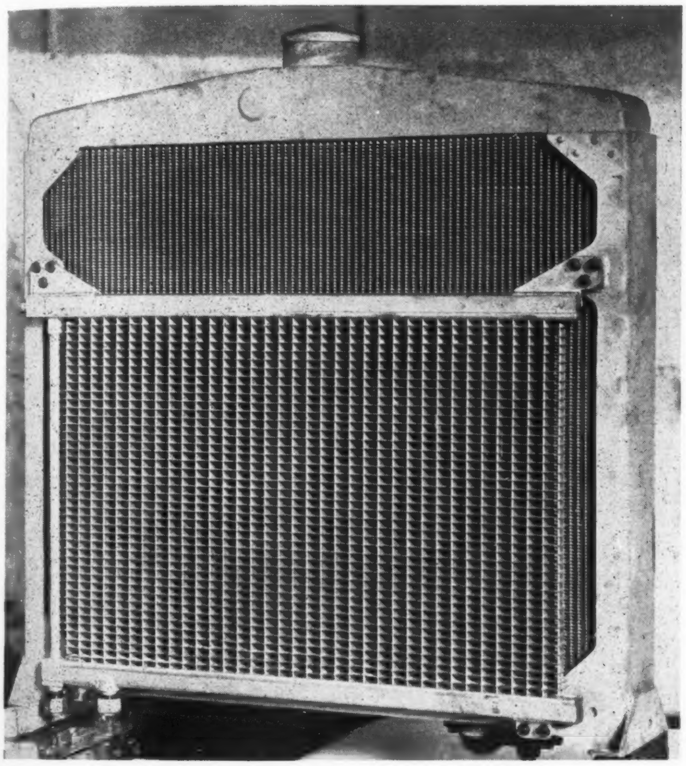


Fig. 3—Oil cooler (in front) for Caterpillar tractor now is made of steel and tests are under way which may lead to use of steel for larger water-cooling radiator in rear

a monthly basis contrasted with actual production of only 40,-000 tons in June, 1942. And the alloy steel supply has been boosted sharply to 1,200,000 tons per month and will reach a peak of 1,340,000 tons within the next few months.

Even so, design engineers face the necessity of specifying NE steels wherever possible for some time to come, since war production continues to be on an ascending scale month-by-month and the available supply of most materials will be readily absorbed. In the last half of 1943 it will be necessary for industry to turn out 60 per cent of the 83 billion-dollar quota for the entire year. The 1944 quota will be even larger.

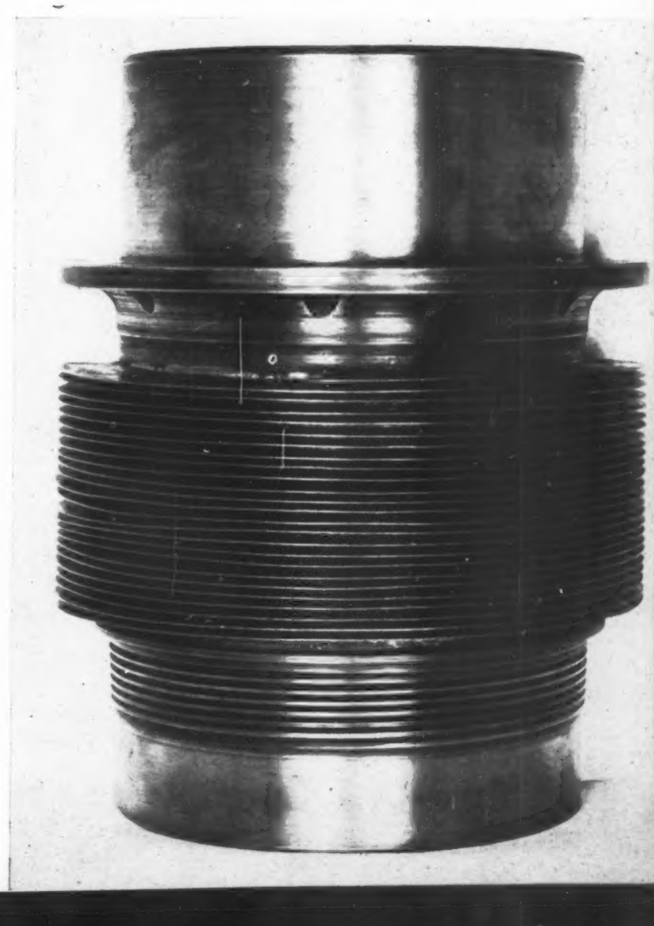
The emergency steels have been working out well for a number of companies, an outstanding example being the Timken-Detroit Axle Co., Detroit. This company recently developed a revolutionary precision forging process for the production of gears, *Fig.* 2, which in itself saves materials. In making a small pinion gear, a blank weighing only .92-pound is needed compared with a 1.65-pound blank when employing conventional methods. The pinions are formed in a series of three forging and two coining operations and require only a minimum amount of final finishing.

The company formerly made two sizes of pinions, using SAE 4620 for the first and SAE 4120 for the second. In making the three sizes now specified for military equipment, the company has substituted NE-8720 and NE-9420. The 8720 steel is found to be a little more susceptible to surface scaling during heating than 4620 and must be handled more rapidly in quenching to obtain the required hardness. The 9400 series steel is proving to be about 85 per cent as good as 4620 from the standpoint of gear life.

Timken engineers are of the opinion that large amounts of the present NE steels will be used after the war, based upon the sound principle that a small quantity of a number of alloys in steel is better than a large quantity of one. This principle has been known for a long time but was not always applied until circumstances compelled its adoption in the form of the NE steels.

Induction heating equipment now being marketed by a number of companies is exerting a strong influence on designers in selecting materials for given parts, two of the tractor builders constituting excellent examples. The Cleveland Tractor Co., Cleveland, has gone over to an induction-hardened medi-

Fig. 4—Cooling fins on this aircraft engine cylinder are copper strip, resulting in savings on alloy steel, machining time and weight per horsepower



um-carbon SAE 1045 steel for a number of parts in its military, industrial and farm tractors from SAE 5120, a chromium steel, which formerly was carburized. SAE 1045 also is used in place of SAE X1020 for rocker arm shafts, link pins and track pins.

The Caterpillar Tractor Co. also is using an induction-hardened SAE 1045 steel in place of SAE 2345, a nickel steel formerly used for tractor final-drive gears. Prior to 1942, 50 tons of nickel alone went into the drive gears for a single model, the Diesel D8. Many other steel substitutions have been made in other tractor parts, including heavy shafts, drawbar pins and push rods.

A list of NE steel substitutions made by the Caterpillar company follows:

Formerly Used	Now Used	Formerly Used	Now Used
SAE 4119	NE-8020	SAE 3115	NE-8620
SAE 4135	NE-8635	SAE 3135	NE-9437
SAE 4140	NE-9437	SAE 2345	NE-8645
SAE 4145	NE-9442	SAE 4820	NE-8827
SAE 4150	NE-9445	SAE 6150	NE-8650

This company also has changed from aluminum to malleable iron for oil pump suction bells, gage panels and fuel-tank caps. Oil and fuel lines have been switched to welded steel tubing from copper. Tubes in the oil cooler shown in *Fig.* 3 now are cold-rolled, welded steel tubing, replacing brass. Fins and headers of these radiators also have been changed over to steel, the fins being solder-dipped to prevent corrosion. The entire surface of the cooler is bonderized and painted. No interior surface material is necessary since the tubes circulate lubricating oil only. Fuel-tank screens have been changed from brass to steel and grommets from rubber to felt.

Considerable success has been encountered by a number of companies with the use of a manganese-sulphur steel, sold on a carbon-steel base, in place of alloy steels. Reed-Prentice Corp., Worcester, Mass., for instance, is using this steel in place of SAE 1045 for machine tool lead screws and finds that wearing qualities have improved and machinability has increased. The same steel also is being used for spline shafts for feed and drive boxes. No redesigning was required in the changeover.

Design engineers who are able to specify bessemer steels as well as rerolled rail steel bars for their products will aid the war effort materially as these are relatively freer than electric furnace and openhearth grades. Nearly 15,000 tons of bessemer and 25,000 tons of rerolled material are produced monthly.

As long as the war is on, copper will continue extremely critical. Although allocations have been increased each quarter by WPB, requirements for the most essential purposes have gone up at a still higher rate. Little expansion in either foreign or domestic production is regarded likely in view of the shortage of labor, equipment and shipping facilities. Two thirds of the available supply goes into brass mill products, one-half of which is in the form of strip for ammunition. Contrary to reports, the steel shell case program has not provided sufficient relief to permit a reduction in production of ammunition brass.

Designers are still urged by the government agencies to eliminate copper from all nonessential applications and specify low-grade, leaded alloys wherever possible in place of high-copper alloys. Electrodeposited coatings should be specified only where they have a definite functional value. Use of the refractory alloys such as silicon, aluminum and beryllium copper, cupro-nickel and nickel silvers also is discouraged since they contain other critical elements and are more difficult to produce than brass.

Copper or copper alloys, according to WPB, should be specified only when necessary to meet the following requirements:

1. Electrical conductivity, but only if use of silver is impracticable
2. Corrosion resistance, provided nonmetallics, lead alloys or coated sheets cannot be made to serve as substitutes
3. Bearings, but only when heavy loads, high torques,

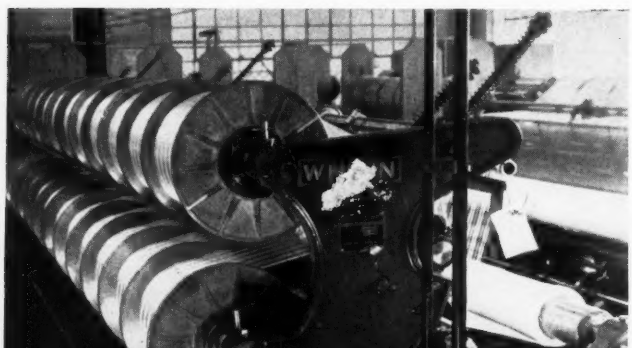


Fig. 5—Cast magnesium heads and magnesium tubing replaced steel and wood in this rayon knitting machine

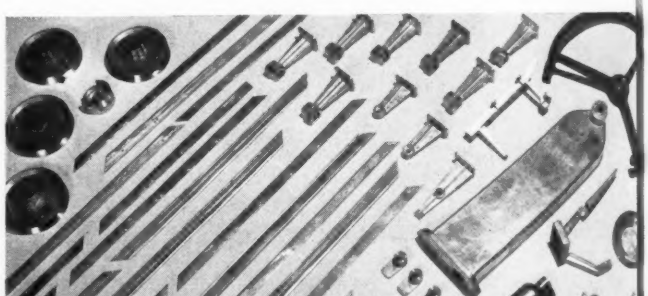


Fig. 6—One hundred and eighteen zinc die-cast parts were used in designing the Link trainer for airplane pilots. The total would have been much larger with machined parts

high operating speeds, or essential safety considerations make lead or silver base or less critical alloys unsuitable

4. Non-sparking, but only when nonmetallics or lead alloys cannot be substituted.

While most substitutions are away from copper, designers will be intrigued with the possibilities of a substitution in "reverse". The aircraft cylinder shown in Fig. 4 makes use of copper in saving alloy steel, increasing horsepower for a given weight and providing greater heat radiation.

Aircraft engine cylinders normally are machined from a single, thick forging but in this design both metal and machining time are saved by replacing the usual fins with copper strip which is wound round the cylinder on a con-

Fig. 7—Unique example of full conversion is this register made by the Standard Register Co. Constructed largely of laminated plywood and hard wood for working parts, it weighs 8 pounds, contrasted with 15 pounds for the replaced model of steel

verted lathe with a fixture which holds the spirals at the proper pitch. Silver alloy brazing wire is wound between the spirals at the same time, with the flux sprayed on following the winding operation. The cylinder then is brazed by placing it in an induction-heating fixture.

Aluminum will be available this year only for the most urgent requirements, which are predominantly for the aircraft industry. The Claimant Agencies were able to obtain only 85 per cent of their needs in the first three quarters of 1943 and the picture in the more distant future is not much brighter due to the expanding aircraft program. The shortage is more particularly one of basic metal rather than lack of fabricating facilities. Labor shortages in the West made it impossible to operate reduction plants at capacity for many weeks after construction had been completed. Plant efficiency also has been reduced by the enforced use of low-grade bauxite.

Up to about two months ago, aluminum extrusions constituted one of the major bottlenecks in the war program with requirements regularly running well ahead of the supply. Since that time, WPB initiated drastic measures to double capacity and orders have gone out to design more extrusions into aircraft. However, designers should make note of the fact that efforts still are being made to encourage use of alloys which may be extruded at high speed and also specify in place of extrusions shapes which may be roll-formed from sheet. The sheet

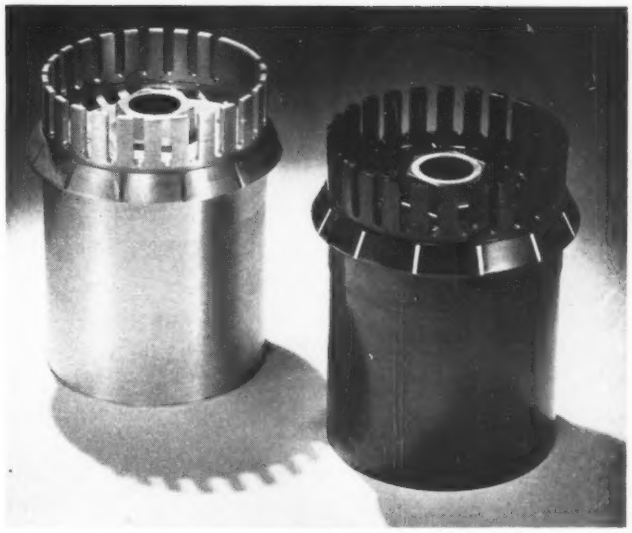


Fig. 8—Speed selector head drum for turret lathe has been converted from aluminum to molded plastic. New drum requires practically no machining

situation has eased for the reason a new continuous mill has gone into operation in the Chicago area. Additional forging capacity now is in operation but it still is insufficient to meet all requirements. Easing of specifications on castings permitting the more extensive use of secondary alloys has relieved an acute situation in this direction.

Although magnesium has been most critical, the situation promises to ease sufficiently in the final half of this year to make this versatile metal available for some new applications. The 1943 supply is estimated at 200,000 tons, of which 40 per cent is earmarked for the aircraft industry, 30 per cent for the chemical warfare service and the balance for export and other outlets associated with the war effort. There appears to be a slight surplus in the latter category and the war agencies no doubt would look with favor upon new semimilitary or military applications.

While strictly military demand for both aluminum and magnesium have limited their normal applications these metals now are serving as permanent substitutes, especially where reduction of worker fatigue and inertia are important factors. For instance, with the rapid increase in employment of women in industry, the use of aluminum housings on small hand tools such as drills and riveting hammers has been immediately reflected in higher production rates because of easier and more rapid handling.

Substitution of Aluminum Reduces Stresses

Similarly, substitution of aluminum for steel for the clicker arm on leather cutting machinery has reduced stresses, resulting in longer bearing and machine life. An important development also has been the use of magnesium tubing and cast magnesium heads, Fig. 5, in the fabrication of sectional warp beams for knitting machines.

Before considering use of zinc in any form, the designer should make certain that no equally satisfactory, less critical material is available and that the application is technically sound. This is especially true with regard to high-grade metal which is needed in the ratio of 30 pounds to 70 pounds of copper in the production of ammunition brass. However, it undoubtedly has been noted that high

grade zinc in the form of die castings has done a particularly outstanding job in the production of many items of military equipment such as signal equipment and the Link trainer, *Fig.* 6 for pilots. Many in industry feel that conservation in use of lower grades of zinc for galvanized coatings has gone too far, especially in view of reports that stocks now are in excess of 75,000 tons or three times the high-grade total.

About 90 per cent of the 5000 tons of cadmium likely to be produced in 1943 will be used as an electroplated coating on steel. Where cadmium is used simply as a protection against corrosion, the war agencies favor the substitution of zinc for the reason that the supply of the former metal will not meet indicated requirements. This substitution is not deemed advisable, however, for close-fitting machine parts where the products of corrosion might interfere with their functioning.

Considerably more success has been encountered re-

Fig. 9—This cooling tower fan blade is laminated plastic

cently with the use of lead as a protective coating for steel. Lead-coated flat rolled steel is available commercially and where this material cannot be utilized, it undoubtedly would be advantageous to investigate present practices in applying lead to specific machine parts. Large stocks of lead are on hand, making the metal freely available for most applications. Antimony, closely associated with lead for many uses, no longer is critical.

Designers should anticipate no relaxation in restrictions on tin either while the war is on or for a period after its termination. While a domestic tin smelter now is operating at Texas City, Tex., its production falls far short of supplying even most restricted war needs. It is estimated that stocks on hand will be reduced by more than 15,000 tons in 1943 and by over 20,000 tons in 1944. However, the supply should last six years unless unforeseen emergency requirements arise. One of the disappointing developments has been the marked reduction in tin produced from secondary materials.

It should be emphasized in looking ahead, however, that the United States no longer will be as dependent upon tin as before the war. The establishment of a complete new electrolytic tinplate industry has materially reduced the major outlet for the metal and engineers think so well of most substitute solders, babbitt and other tin-base bearing metals that a return to high-tin alloys is regarded as unlikely.

Arsenic Babbitt Bearings Perform Well

The new so-called arsenic babbitts with 80 per cent lead, 16 per cent antimony and 1 to 3 per cent arsenic are reported to be performing as well as the tin-base babbitts, one of which contains 85 per cent tin, 10 per cent antimony and 5 per cent lead and another 8 per cent tin, 14 per cent antimony and 78 per cent lead. Furthermore, bearings made with the substitute material are less costly.

In discussing bearings it also might be well to point out the progressive reduction in use of bronze bearing materials, the tendency being to cut down on the thickness of the liner and increase the strength of the shell. Engineers also are looking with increased favor upon powder-metal bearings which have a faculty for retaining oil so that if lubrication is faulty the bearing will be self-lubricating for a certain length of time.

As for the solders, it should be observed that bismuth, used in substitutes for tin-lead solders, is becoming more critical. In fact, elimination of bismuth in general purpose solders and babbitts is being considered. Silver is being produced in ample volume for use in bearings, silver solders and brazing alloys.

Ruthenium Substitutes for Iridium

In briefly reviewing the situation with respect to rare metals, note should be made of the fact that makers of iridium electrical contact points have been asked to substitute ruthenium wherever possible. Platinum, osmium, palladium and rhodium, also used for electrical purposes, are not particularly critical.

The nonmetallics have assumed increasing importanace in the emergency, not only as excellent substitutes but in new, permanent applications. Capacity for production of synthetic resins is close to 200,000 tons annually and a much larger tonnage of finished products may be turned out through their use as binders for plywood, wood flour, paper, asbestos and the like. The resins, of course, are also being formed into an endless variety of products, *Fig.* 7, among the latest being seamless tubing complete with fittings which make it readily adaptable for many machine applications. The manufacturers of several types of synthetic rubber are now readying more extensive lines of mechanical goods for the market and many of these are superior to the natural product, especially when subjected to the corrosive action of oils and lubricants.

The drum, *Fig.* 8, for the head for determining machine cutting speeds on Warner & Swasey turret lathes may be cited as an example of evolution in design made necessary

(*Concluded on Page* 230)

BOLT FAILURE as Affected by Tightening*

By J. O. Almen
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FATIGUE of bolts and studs has been widely discussed and the improvements resulting from reducing the diameter of the bolt body and from pressure rolling of the threads have been adequately recorded. Insufficient attention, however, has been given to fatigue vulnerability due to insufficient bolt tightness. A bolt or stud should be tightened to a load exceeding the maximum working load. When properly tightened against rigid members, a bolt cannot fatigue because there can be no change in stress and the bolt load therefore is static even though the load applied to the bolted member may change at high frequency from zero to maximum. The foregoing rule must, however, be used with caution because all bolted members are elastic in some degree and the design of the bolted members may be such that the applied load is greater than can possibly be supported by the bolt.

Excessive Stress Occurs

An exaggerated case of this is shown at left of *Fig.* 1 in which the bolts are excessively stressed in tension and in bending because the distance from the bolt to the point *C* is small; since the bolted parts tend to bend about the point *C* as a fulcrum, the tension and bending loads in the bolts are great. *Fig.* 1 shows, right, an improved design in which the fulcrum point *C* is farther removed from the bolt and, therefore, the tension and bending loads are reduced. In this example fatigue failure of one member may occur due to faulty design of another member, as is frequently encountered in practice. It must be remembered that, like an aching tooth, the one that seems to hurt is not necessarily the one that should be pulled.

If the right-hand bolt should fail by fatigue the failure could still not be charged to insufficient bolt strength because, as stated above, if the initial bolt tension is less than the applied load the stress range under repeated loads is increased. If the bolt were tightened just enough to bring the surfaces into contact without appreciable tension, then under alternating stress the stress range would be from zero to maximum and fatigue failure could only be avoided by greatly increasing the bolt strength. As the initial bolt tension is increased the stress range is decreased until it approaches zero, when the initial bolt tension is equal to or greater than the maximum working load. This illustrates a case in which a bolt fatigue failure is not the fault of either bolt strength or of design but is chargeable to bad assembly practice.

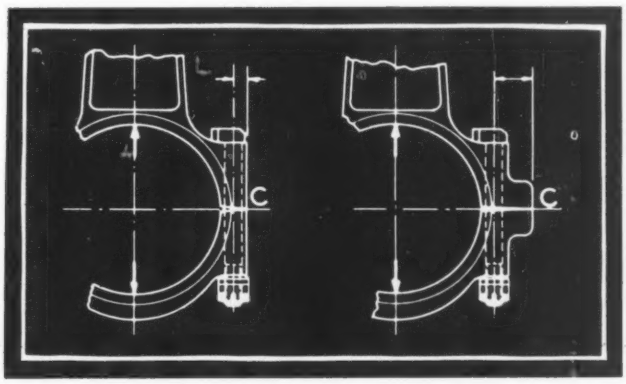
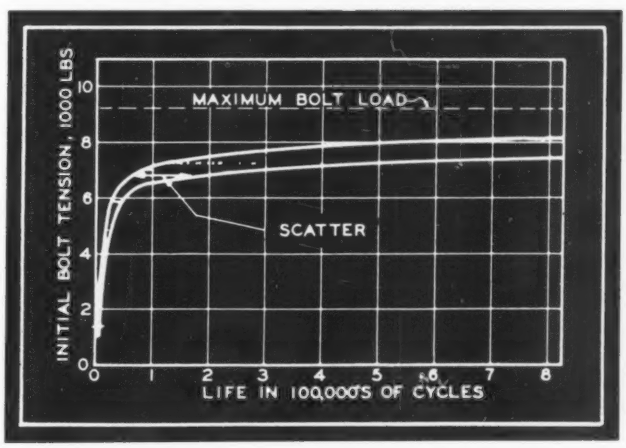


Fig. 1—Tensile and bending loads on bolt at left are excessive due to short moment arm. Condition is alleviated by the improved design at right

Fig. 2—Tests on the effect of initial tension on bolt life show importance of adequate tightening



Vulnerability to fatigue as a function of bolt tightness is shown in *Fig.* 2. In these tests all bolts were sub-

*From a paper presented at a meeting of Detroit chapter, A.S.M.

jected to a cyclic tension load of 9210 pounds but were tightened to initial tensions of 1420, 7220 and 8420 pounds. Fifteen bolts were tested in each of the three lower groups of the graph in order to establish partially the scatter band for this kind of specimen. Only two bolts were tested in which the initial tension was 8420 pounds, one of which failed after 4,650,000 stress cycles and the second did not fail after ten million stress cycles. These are not shown on the graph because they would compress the stress scale to undesirable proportions. The bolts used in these tests were ⅜-inch in diameter accurately dimensioned and finished. The threads were U. S. form, 24 threads per inch and ground to close limits.

Stress Range Affects Durability

The stress range to which these bolts were subjected is the difference between the initial load and the maximum operating load and since it is known that fatigue durability is increased as the stress range is decreased, results of the order that were obtained from these tests as shown in the chart are to be expected. All failures occurred in the threads except in a few cases in which the threads were rolled in a manner to pre-stress the roots of the threads in compression. In these rolled threaded

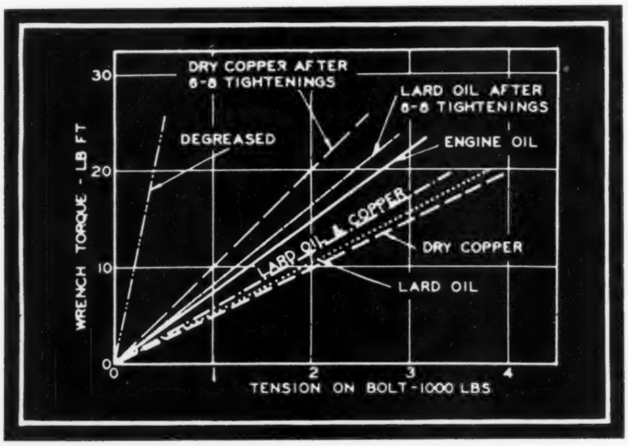


Fig. 3—Relation between wrench torque and bolt tension is greatly affected by lubricant and surface condition

bolts the fatigue durability of the threads was increased sufficiently to cause failure in the bolt shanks. When the surfaces of the bolt shanks were also compression prestressed by peening, the failures were again transferred to the threads but, of course, at prolonged durability. These tests, therefore, also show that the fatigue durability of cut and ground screw threads can be increased by rolling and indicate that compression pre-stressing of the surface of pure tension members is effective in increasing their fatigue strength.

It is evident that the fatigue strength of bolts and studs stressed in tension is dependent upon the initial tension applied by the nut and upon the elasticity of the bolted members. Therefore washers, lock washers, gaskets and other devices that add to the elasticity of the bolted assembly are fatigue hazards and should be avoided if possible.

Initial tension applied by the nut is difficult to determine. Measurement of torque applied to the wrench is unreliable because of the variability of friction. *Fig. 3* records tension measurements plotted against wrench torque in pounds-feet for 5/16-inch diameter bolts having 24 threads per inch. It will be seen that the bolt tension varied as much as ten to one for constant wrench torque depending upon the lubricant used. The only sure way to determine initial tension is to measure the elongation of the bolt.

Bolt Loads Analyzed

IN THE foregoing article the detrimental effects of insufficient tightening of bolts and of too much elasticity in the connected members were emphasized. Supplementing this article, the following quantitative analysis of the effect of bolt and member stiffness sheds further light on the problem of bolt loading.

When an external load W is applied to a bolt which already has been tightened to an initial tension W_0 the bolt tension can increase only in proportion to the amount that the bolt stretches. Assuming the stretch is Δ and the bolt stiffness is k_b (pounds per inch of stretch), then the additional bolt load is $W_b = k_b\Delta$. At the same time the bolted members which were in compression also deform by the amount Δ. If their stiffness is k_m, the corresponding reduction in the compressive load is $W_m = k_m\Delta$. The total applied load W is the sum of the increase in bolt tension and the decrease in member compression, or $W = W_b + W_m = k_b\Delta + k_m\Delta$, provided the initial tension is not less than W_m. From this relation, $\Delta = W/(k_b + k_m)$ and the increase in bolt tension is $W_b = k_b\Delta = k_bW/(k_b + k_m) = W/(1 + k_m/k_b)$.

Example Illustrates Effect of Stiffness

As an example the case may be considered where the members are three times as stiff as the bolt ($k_m/k_b = 3$). The added bolt load is $W_b = W/(1+3) = W/4$. The decrease in member compression is $W_m = W - W_b = 3W/4$, therefore the initial tightening load should be not less than three-fourths of the applied load for this case. Assuming the applied load is 100 pounds the tightening load should be at least 75 pounds. Under this condition the members would be entirely relieved of compression when the load was applied. To insure the members always being in compression it is a safe rule to make the initial load slightly greater than the applied load. Assuming an initial tightening to 110 pounds, the total bolt tension after application of the 100-pound external load would be $110 + 25 = 135$ pounds, while the net compression in the members would be $110 - 75 = 35$ pounds. When the load is periodically applied the bolt therefore goes through a load cycle from 110 pounds minimum to 135 pounds maximum.

Effect of increased member stiffness is shown by assuming $k_m/k_b = 9$, say, when the added bolt load would be only $W/10$ or 10 pounds, giving a maximum of 120 pounds with an initial load of 110 pounds as before. The stress range in fatigue loading is therefore less when the members are stiffer.

How SUPERCHARGERS Aid High-Altitude Flight*

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***Fig. 1**—Top—Two-stage supercharging of Flying Fortress engines contributes to exceptional high-altitude performance—Boeing Aircraft photo, courtesy OWI*

BASIC function of supercharging—to increase the intake manifold air pressure in an engine—has two immediate results: The density of the charge forced into each cylinder is increased, and a favorable excess or differential is maintained in the intake manifold pressure with respect to the exhaust back-pressure. These two effects are utilized to accomplish the primary objectives of supercharging aircraft engines:

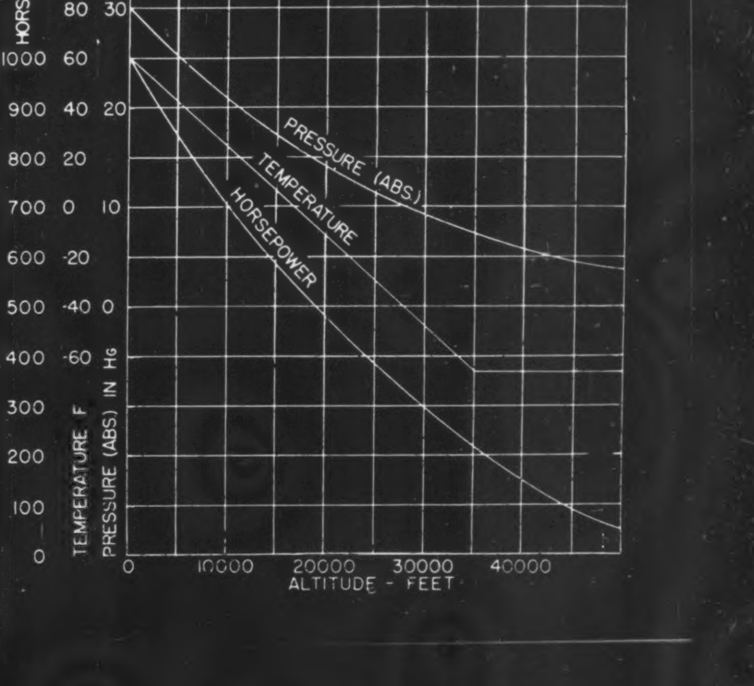
1. To maintain full rated power at altitude
2. To increase power at sea-level (ground boosting)
3. To improve fuel economy for cruising.

A further beneficial effect of supercharging is to promote fuel vaporization and uniform distribution of mixture to the individual cylinders.

In order to appreciate the nature of the problem it is first necessary to take account of a few elementary relationships underlying the performance of present-day aircraft engines. Other things being constant, the indicated horsepower developed in an engine is directly proportional to the mass rate of air consumption (pounds per minute). For a given speed this means that power is

*Abstract of a paper presented at the recent semiannual meeting of the A.S.M.E. in Los Angeles.

***Fig. 2**—Below—How atmospheric pressure and temperature vary with altitude. Effect on horsepower of an unsupercharged engine also is shown*



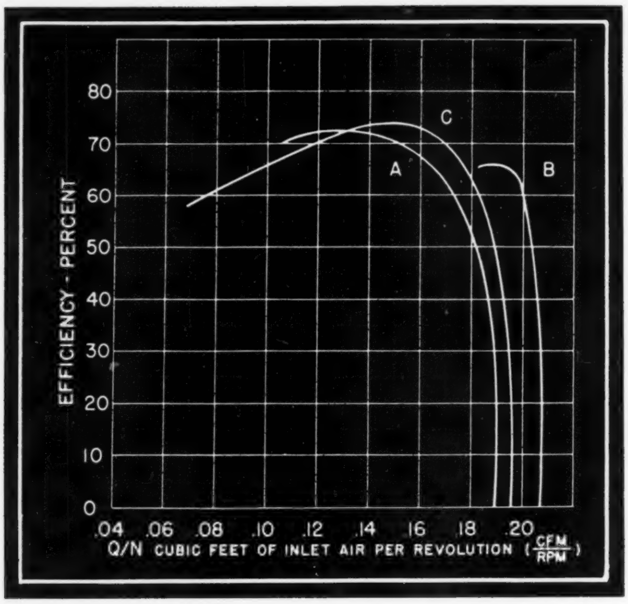


Fig. 3—Typical range curves for superchargers show narrow range of operation for which efficiency is satisfactorily high

proportional to the density of the air, which is a function of its pressure and temperature. As a fairly close approximation(1)*, the brake horsepower of an engine varies directly with the absolute pressure, p, and inversely as the square root of the absolute temperature, T, of the carburetor inlet air, as expressed in the formula:

$$\frac{bhp_0}{bhp_1}=\frac{p_0}{p_1}\sqrt{\frac{T_1}{T_0}} \quad \text{(1)}$$

where the subscripts 0 and 1 refer to standard sea-level and altitude conditions, respectively.

How these factors affect the altitude performance of a modern unsupercharged aircraft engine, using standard N.A.C.A. atmospheric data(2), is shown in *Fig.* 2. A glance at this should suffice to indicate the occasion for supercharging military airplane engines, where it is so vitally important for the engine to be capable of developing full power at any level within its altitude range.

By the use of a supercharger an engine which gave less than 30 per cent of its full power at 30,000 feet can deliver 100 per cent power from sea level to that altitude, and beyond. Further, a plane having a speed of 250 miles per hour at sea level may attain a speed of 350 miles per hour at 30,000 feet for the same power and time-rate of fuel consumption. High-altitude flight under these conditions, therefore, yields marked improvements in range and economy.

Superchargers can also be used effectively for "ground boosting", that is, increasing the sea-level power of the engine, since the indicated mean effective pressure (I.M.E.P.) and horsepower are proportional to the intake manifold absolute pressure (M.A.P.). With a moderate allowance for friction and accessory power loss in the engine, the brake mean effective pressure (B.M.E.P.) and horsepower likewise increase almost linearly with the M.A.P. It is entirely practicable, for example, to increase the output of an unsupercharged engine from 500 to 800 horsepower by boosting the M.A.P. from 29 to 40 inches of mercury, absolute, without changing the speed. This extra power is frequently of great value in facilitating take-off.

For cruising, power can be reduced either by reducing the speed or the B.M.E.P., or both. If the propeller efficiency is not impaired, it is more economical to reduce power by reducing the speed, keeping the B.M.E.P. as high as practicable. This is a consequence of the reduction in engine friction losses at the lower speeds.

However, a serious obstacle to the maintenance of high manifold pressures for cruising is the limited range of satisfactory operation of superchargers now in common use, as indicated in *Fig.* 3. The value Q represents the volume flow of air supplied to the engine and is proportional to the engine power; N is the supercharger speed, which is the chief factor in determining the pressure (M.A.P.) supplied to the intake manifold. The useful operating range of a supercharger refers to the spread between the lower value of Q/N, usually established by the incidence of surging or instability, and the upper value beyond which the efficiency falls off rapidly. Curve *A* is representative of the modern centrifugal supercharger and indicates the difficulty of attempting to reduce Q for low power outputs while maintaining N at a relatively high level in order to employ high manifold pressures.

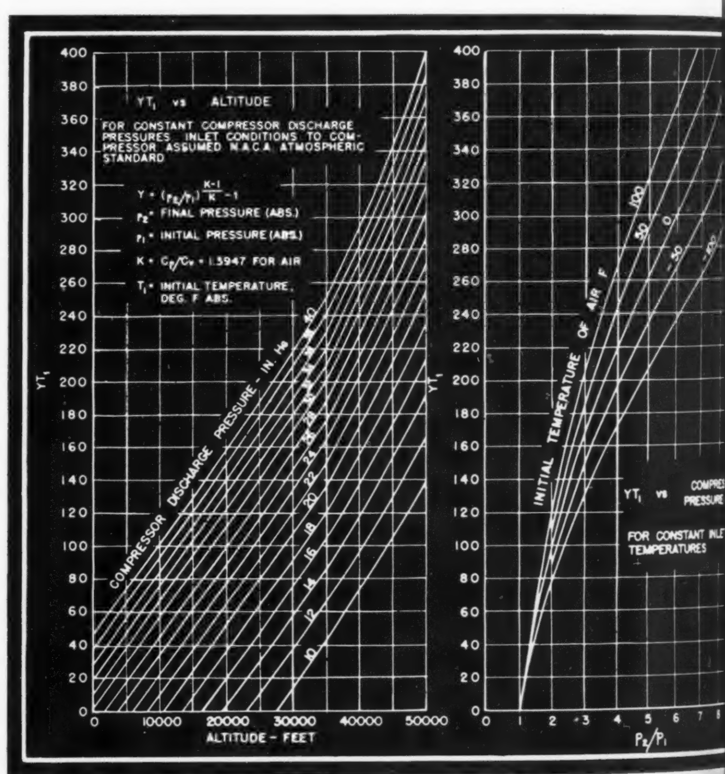


Fig. 4—Horsepower required to compress air is a function of YT₁ which in turn is affected by pressure ratio and initial temperature (right). Variation of this function with altitude and discharge pressure is shown by the series of curves at left

*References in parentheses are listed at end of article.

In addition to the fact that this tends to push the value of Q/N into the unstable region beyond the left end of the curve, a flexible speed control is required to allow N to be maintained when the engine speed is reduced to less than half of its maximum rating.

Present-day aircraft engines consume about .12-pound of air per minute per horsepower delivered. The power required to compress this air is determined chiefly by the ratio of the final to the initial absolute pressures p_2/p_1, and the absolute temperature T_1 degrees Fahr. absolute, of the inlet air. The theoretical power for isentropic compression (3) is given by:

$$hp = .00573\ Y\ T_1 \text{ per pound of air per minute} \ldots\ldots (2)$$

$$\text{where } Y = \left(\frac{p_2}{p_1}\right)^{.283} - 1$$

Values of the function YT_1 are given in *Fig.* 4. The curves on the left are based on the assumption that p_1 and T_1 correspond to the standard N.A.C.A. atmospheric values of *Fig.* 2 for each altitude. Curves on the right give the same function for the particular pressure ratio and inlet temperature involved in any case. To obtain the compressor shaft input power the values obtained from Equation 2 must be divided by the efficiency.

In many practical cases p_1 may be assumed to be equal to the prevailing atmospheric pressure in flight, since the pressure drop in the upstream induction system is roughly balanced by the "ram" or kinetic pressure of the slip-stream impinging upon the air scoop. For the typical internal geared supercharger p_2 is the manifold pressure. Assuming a compressor efficiency of 70 per cent, the supercharger power required to develop 1000 horsepower at 10,000 feet with a manifold pressure of 32 inches Hg can then be obtained from *Fig.* 4 and Equation 2 as follows: $YT_1 = 65$, air consumption $= .12 \times 1000 = 120$ pounds per minute, and horsepower $= (.00573 \times 65 \times 120)/(.7) = 63.8$. This must again be divided by the mechanical efficiency of the drive train to obtain the power taken from the engine crankshaft. The pressure ratio in this case is 1.55. For comparison, a supercharger to maintain 40 inches of manifold pressure at 25,000 feet would have to develop a pressure ratio of 3.6, which would require an input of about 154 horsepower for a 1000-horsepower engine.

Unfortunately, air cannot be compressed even with the perfect efficiency of an isentropic process, without a significant rise in temperature, unless the heat of compression is removed by cooling during the process. In aircraft superchargers it is not practicable to effect any appreciable amount of cooling in the compressor itself; in fact the surface area is so small relative to mass flow of air that the process is substantially adiabatic (and polytropic).

For an adiabatic process the minimum temperature rise occurs in the case of reversible, isentropic compression, that is, when the compressor efficiency is 100 per cent. In this case the ideal temperature rise is numerically equal to the function YT_1 of *Fig.* 4 or $T_2 - T_1 = YT_1$, and the actual temperature rise in any case is this value divided by the compressor efficiency.

To determine the temperature rise of the air in the preceding examples of supercharger power, for the 32-inch manifold pressure at 10,000 feet, $T_2 - T_1 = YT_1/0.7 = 65/0.7 = 93$ degrees Fahr., and for the 40-inch pressure at 25,000 feet, $T_2 - T_1 = 188/0.7 = 269$ degrees Fahr.

With an inlet air temperature of —30 degrees Fahr., the manifold temperature in the latter case would be 239 degrees Fahr. This would be intolerably high, and in practice the total pressure ratio of 3.6 involved in this case would be divided between two stages of compression with an intercooler between them, as indicated in *Fig.* 5. Under typical conditions with this arrangement the manifold temperature would be reduced to 134 degrees Fahr.

There are at least three reasons for limiting the manifold temperature to the lowest practicable level:

1. To avoid detonation or "knocking" in the engine, which has such a serious effect upon engine life as to constitute a major limitation to the allowable power rating. As the mixture temperature increases, a richer mixture or fuel of higher octane rating must be employed to maintain constant intensity of detonation.
2. To increase the density of the charge supplied to each cylinder, thereby increasing the power developed, Equation 1.
3. To reduce the combustion and exhaust temperatures of the engine.

These are promoted by improving the efficiency of the supercharger, which reduces the temperature rise during compression. In addition, any improvement in first-stage

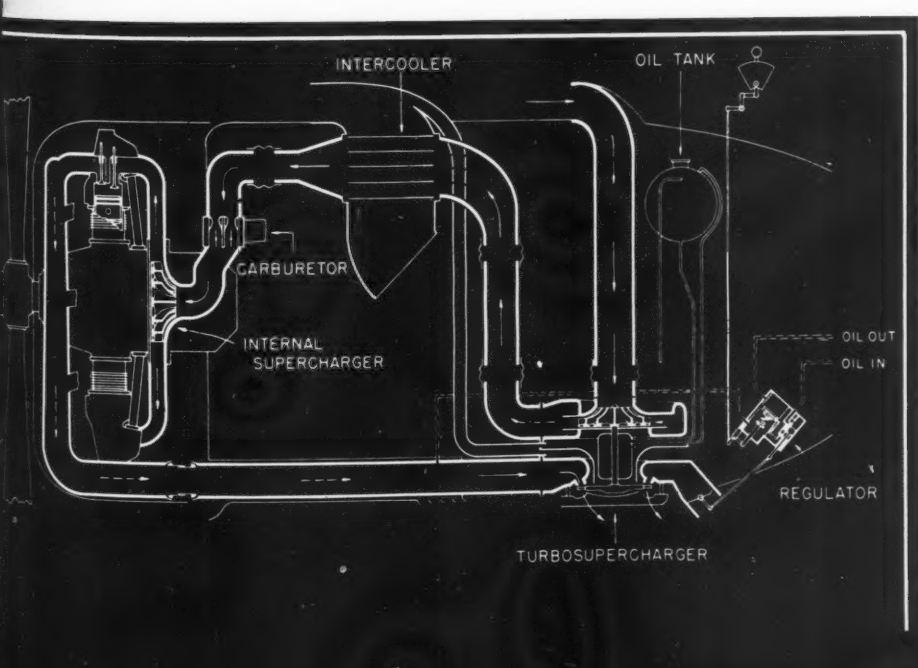


Fig. 5 — Schematic diagram shows arrangement of a turbo-supercharged airplane power-plant with internal supercharger geared to engine

efficiency reduces the work of compression in a second stage by reducing the value of T_1 in Equation 2.

On the other hand, these same results can be promoted by increasing the effectiveness of the intercooler. In either case higher efficiency or effectiveness inevitably results in greater size and weight. Optimum installation design therefore calls for a careful compromise of these factors to obtain best overall performance, with due regard for total drag, power and supercharger operating speed.

Principal elements of the centrifugal compressor are a multivaned impeller and an annular diffuser enclosed in a suitable casing, which includes a collector or scroll surrounding the diffuser. The impeller is driven at speeds of the order of 20,000 to 30,000 revolutions per minute, with peripheral velocities or tip speeds commonly reaching 1100 or 1200 feet per second. The impeller imparts energy to the air principally by giving it violent peripheral and angular acceleration, roughly half of this energy appearing as static pressure rise and half as kinetic energy at the exit(4). The function of the diffuser is to convert as much as possible of the velocity of the high-speed air streams into further increase in the static pressure, by a process of efficient deceleration.

Because of its high speed of operation the centrifugal supercharger has an exceptionally high volumetric capacity, making it especially well adapted to use in aircraft.

In selecting a centrifugal compressor for a given supercharger application the following factors must be taken into account:

1. Capacity	5. Impeller Speed
2. Max. pressure ratio	6. Mechanical ruggedness
3. Efficiency	7. Size
4. Range	8. Weight

Capacity, usually expressed in pounds of air per minute, is governed primarily by impeller diameter, D, and revolutions per minute, N. For a series of *similar* machines the volumetric flow Q (cubic feet per minute) is proportional to ND^3, so that all sizes of such a series should have the same value of Q/ND^3. The basic capacity of the design, or "characteristic flow", which Q/ND^3 represents can be increased by increasing the depth of the impeller passages or the *relative* area of the inlet. Any considerable departure from optimum proportions, however, will impair the efficiency.

As previously indicated, a supercharger must deliver .12 pounds of air per minute per engine horsepower. Volume flow at any altitude is obtained by dividing mass flow by the density, to obtain the value of Q. The machine must then be selected or adjusted to bring the rated value of Q/N to the appropriate point near the peak of the characteristic curve (*Fig.* 3). In practice, final adjustment of the capacity of the compressor is accomplished by suitably changing the angle and flow area of the diffusers (this is done by changing the entire diffuser, which is nonadjustable). The volume flow required to maintain a given capacity increases rapidly with altitude, becoming 6.6 times as great at 50,000 feet as at sea level.

Maximum pressure ratio is an important criterion of the excellence of a given design, as it fixes the useful ceiling of the machine. The pressure-producing capacity of a centrifugal compressor is measured by the so-called "pressure coefficient":

$$\eta = \frac{6083\, Y T_1}{V^2} \qquad (3)$$

where V is the impeller tip speed, feet per second. This coefficient compares the minimum energy theoretically required to compress the air between the pressures involved in Y with the energy actually imparted to the air by the impeller, which is proportional to V^2. From Equation 3 the necessary operating speed to deliver the required pressure ratio can be obtained, the value of YT_1 being taken from *Fig.* 4.

In general, pressure coefficients for impellers with straight radial blades vary with Q/N in about the same manner as the efficiency (e_t), as shown by the curves of *Fig.* 3. Although Equation 3 does not apply rigorously to impellers having curved blades, it nevertheless may be said that forwardly-curved blades produce relatively high pressure coefficients, in that a lower tip speed is required for the same pressure ratio. This is secured, however, at the expense of a considerable reduction in stability and range of operation. Conversely, impellers with backwardly-curved blades commonly show higher efficiencies but lower pressure ratios for the same tip speed than

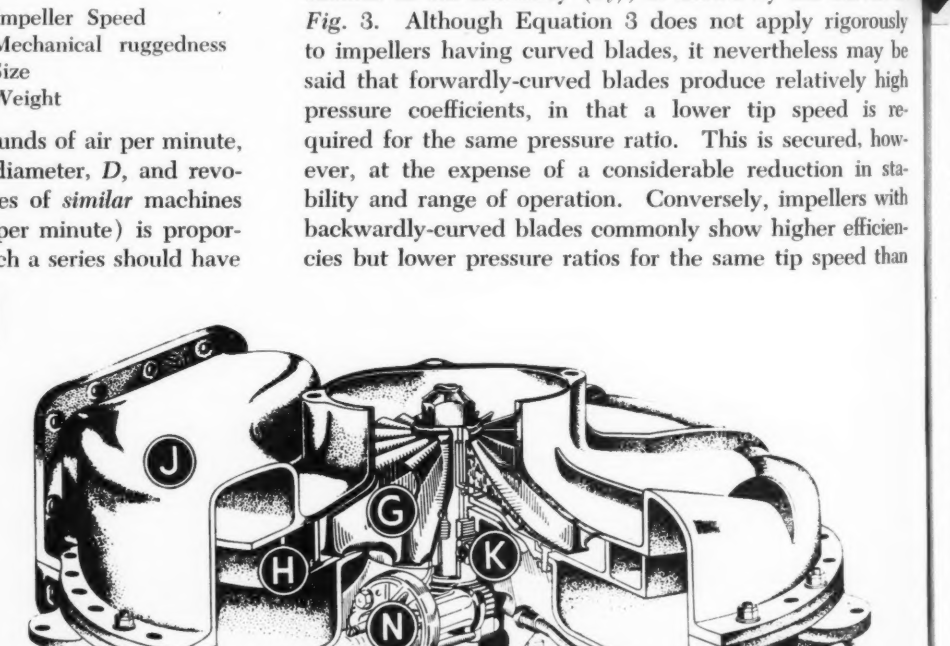


Fig. 6—Cutaway view of turbosupercharger unit shows turbine element below, compressor element above

the straight-bladed type.

Substituting the value of Y in Equation 3 it can be shown that

$$\frac{p_2}{p_1} = \left(1 + \frac{\eta V^2}{6083\ T_1}\right)^{3.534} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

The maximum pressure ratio which a given machine can develop is usually limited by the allowable stresses due to centrifugal force, which is proportional to the square of the speed. But if this is not the limiting factor there is a general tendency for the pressure coefficient and range to decrease at high speeds, as indicated by the change from curve A to curve B in *Fig.* 3. The ultimate ceiling may be reached at the point where the value of η collapses so that there is no further increase in pressure with speed.

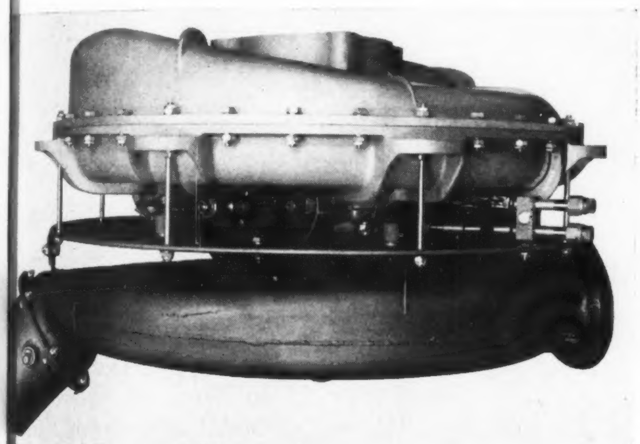


Fig. 7—Side view of turbo-supercharger. At lower left is waste gate which by-passes excess gas to the atmosphere

Efficiency of a centrifugal compressor depends almost entirely upon how successfully the air is introduced into the impeller and conducted through the passages of the machine without shock, flow-separation or turbulence. This sounds much simpler than it actually is. The air entering the impeller is given a sudden, terrific acceleration, sometimes reaching a figure of 6,000,000 feet per second per second, or 186,000 times gravity. The internal air velocities are commonly of the order of 1000 to 1200 feet per second at the diffuser entrance. The latter figure would exceed the velocity of sound (about 1088 feet per second at 32 degrees Fahr.) except for the fact that this velocity increases in proportion to the square root of the absolute temperature, which increases considerably with the speed. At these speeds the air particles tend to travel through the flow passages like rifle bullets, so that it is difficult to make them turn corners and fill the varying flow sections without separation.

At the higher air velocities associated with relatively high volume flow and tip speeds the performance of the compressor is usually limited by the occurrence of shock waves manifested by sudden changes in pressure with high energy losses, as the velocity at the diffuser entrance approaches the sonic or acoustic value. The tendency of these shock losses to occur is measured by the so-called "Mach number", which is simply the ratio of the air velocity, u, to the velocity of sound, C, at the local temperature (or any convenient reference temperature). Since the critical value of u is determined by and is near-

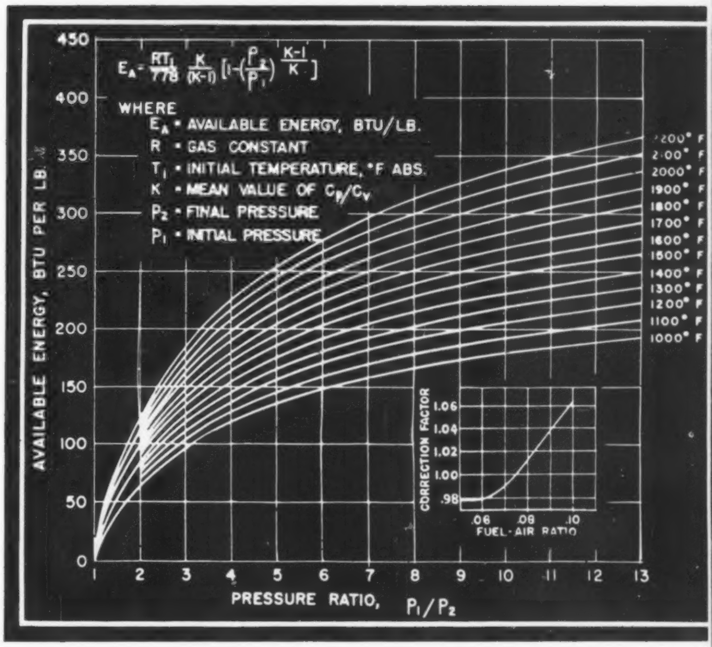


Fig. 8—Available energy in exhaust for fuel-air ratio .075
For other mixtures multiply by correction factor

ly equal to the tip speed, V, and since $C = 49\ (T)^{1/2}$ it is convenient to express the local Mach number as $M = V/49(T)^{1/2}$. Unfortunately, the local value of T at the diffuser inlet is usually not known, but for a given value of V it is determined by the inlet air temperature, T_1, so that it is still more convenient to use a distorted Mach number $M_1 = V/49(T_1)^{1/2}$ in comparing the basic efficiencies of different compressors. This means that all of the values of efficiency versus Q/N should fall on the same curve, as A in *Fig.* 3, for a constant value of M_1 no matter how the speed and inlet temperature may be varied. A higher value of M_1 would tend to reduce the performance in the direction of curve B.

Since efficiency, pressure coefficient and range are thus impaired by the high speeds and low temperatures required for high-altitude operation it is customary to employ two stages of compression when the total pressure ratio is greater than about two or three to one. However, there is always a strong incentive to develop single-stage designs to yield the utmost in pressure ratio without too great sacrifices in performance, for reasons of mechanical simplicity, weight and size.

Significance of the problem of operating range has already been indicated above. The problem is becoming progressively more acute as the spread between maximum and minimum engine power increases and as higher altitudes call for higher supercharger speeds. One expedient for increasing range is the use of a ring-type or vaneless diffuser, which gives the type of performance represented by curve C of *Fig.* 3. However, the penalty is usually a substantial increase in size(5) or else a loss in efficiency and pressure coefficient(6).

Speed, mechanical ruggedness and weight are closely interrelated factors. The high speeds at which centrifugal superchargers operate not only set up high centrifugal stresses in the impeller but also generate vigorous vibrations in all parts of the machine, due to resonance effects

(*Continued on Page* 214)

Drafting Standards Revised

By F. G. Higbee*
University of Iowa

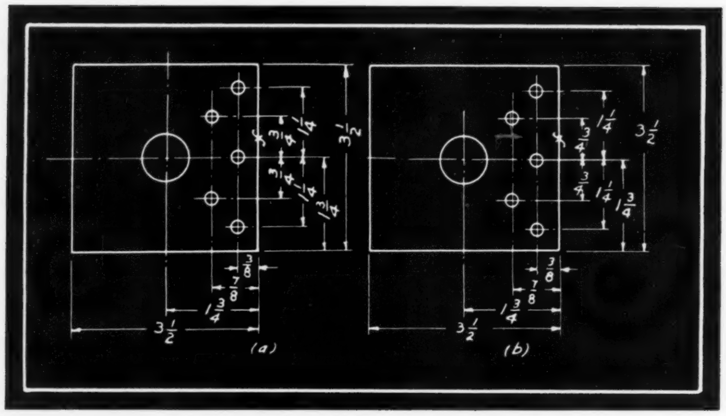
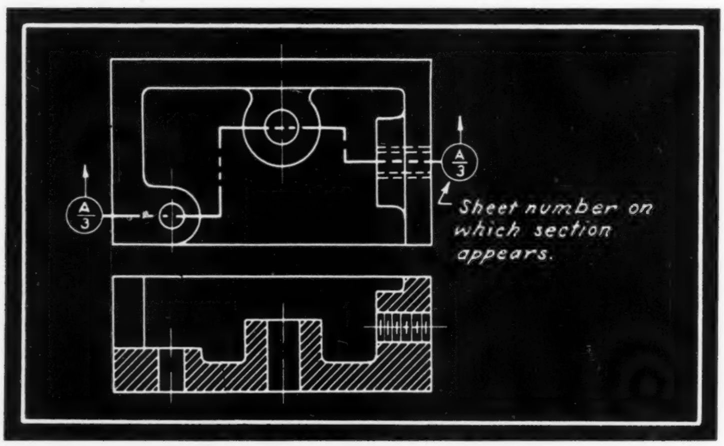


Fig. 1—New standard will approve unidirectional dimensioning (right) which is advantageous for large drawings, as well as the more conventional aligned system (left)

Fig. 2—Letters identifying section are placed in circle as numerator of fraction, denominator showing sheet on which section appears



ALTHOUGH the American Standard for *Drawings and Drafting Room Practice* (A.S.A.—Z14.1) which was approved by the American Standards Association in May, 1935 was well received and widely approved, it has for some time been felt that many comparatively new techniques and practices should now be reviewed and incorporated into a revised standard if found to be sound and deserving of recognition. In consequence of this belief, such a revision has been proposed by the sectional committee and is now being studied, reviewed, and edited in preparation for approval.

The principal additions to the 1935 standard which it is believed will be of interest and value are:

1. Approval of the "unidirectional" system of dimensioning, *Fig.* 1. Since this practice has become common in aircraft and automotive drafting the standards now recognize this system, which places all dimension figures to read from one side of the sheet
2. A standardization of names for the several auxiliary views in general use
3. A more complete statement of structural dimensioning practice
4. New methods for identifying sections by the use of lines, letters and numbers (*Fig.* 2)
5. An enlargement of the present material on dimensions, dimensioning practices and limit dimensioning
6. A reconsideration and study of methods for the indication of finish and of surface quality. This matter is still under survey and study
7. General editorial revisions of both text and illustrations to clarify and amplify points about which inquiry has been made.

At the time the first standard was being considered, all sorts of slang, conventional representations, idiomatic expressions, colloquialisms, and ungrammatical forms were being used with no authority except that each concern found a need for some way of graphically expressing an idea and, in the absence of a standard, had devised their own.

The principles on which a standard is accepted and established are common-sense rules. The fact that a large number do it this way or that way is of no great influence. Every representation is scrutinized to see if it satisfies certain conditions: First and foremost, it must clearly indicate its meaning without ambiguity; second, it must be in accord with generally accepted theories of graphic representation; third, it must be as simple as possible so as to be economical to use; and fourth, when all three foregoing conditions are satisfied, it must be practical.

It was not expected when the first standard was published in 1935 that a universal revision of individual practices would take place at once. There has been, however, a gratifying response. The published literature, both textbooks and technical magazines, show the influence of these standards, and scores of letters approving the proposed revisions indicate how widely industry is already in agreement. There is no question that year by year the language of drawing approaches steadily that ideal condition where uniformity is universal.

*Chairman of the subcommittee on revision of the standards for Drawings and Drafting Room Practice of the American Standards association.

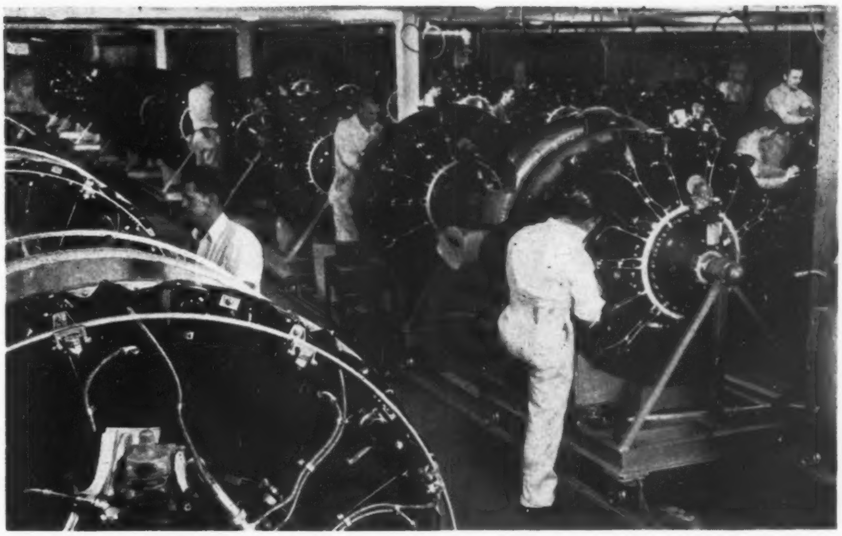


Fig. 1 — Assembly - line trucks in aircraft engine plant utilize straight-fork casters with grooved wheels. Track is inverted angle iron, reducing hazard of obstructions

Selecting Wheels and Casters

Part II

By John W. Greve

BECAUSE there are no hard and fast rules or formulas which govern the selection of a caster for a particular application, experience or comparison with similar applications is necessary, as well as knowledge of standard types available. Certain basic factors, however, are always present for consideration. These are briefly:

1. Value of equipment
2. Weight of load
3. Floor conditions
4. Movement desired
5. Mounting method

Evaluation of these five factors leads to a selection which may be modified within limits to achieve a satisfactory application of standard casters without the delay and costs involved with special designs.

Value of equipment to which casters are applied often escapes the primary attention it deserves. Too frequently casters are regarded as appendages to be added after the equipment is designed instead of providing for them in the original layout of the machine.

Delicate loads require shock-absorbing wheel treads, such as leather or rubber to reduce vibration and shock. Also the diameter of the wheel should be larger than otherwise would be used. Many a fine unit, embodying dozens of man-hours of work and hundreds of dollars worth of delicate parts, is mounted on a set of casters

Fig. 2—Large, heavy refrigerating unit utilizes eight eight-inch phenolic plastic swivel casters. Unit is truck-drawn for long hauls and manually pushed for positioning in close quarters

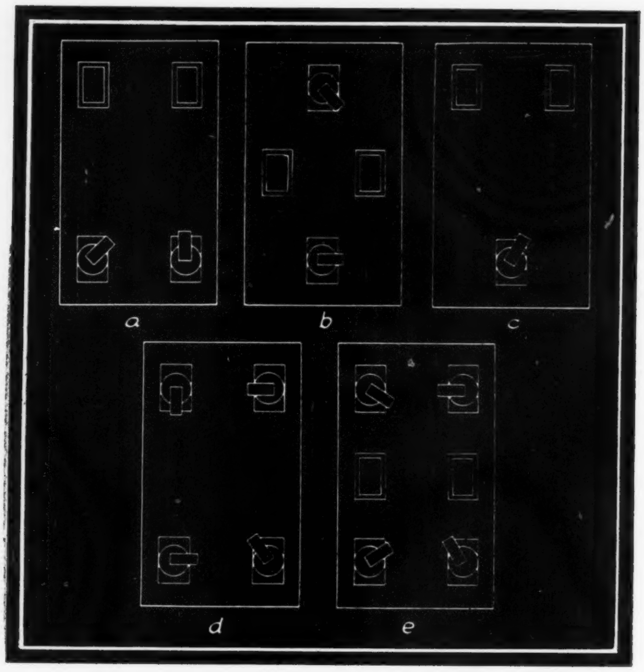


Fig. 3—Typical caster arrangements for general conditions involving long-distance travel or movement in close quarters: (a) For long distances and easy steering; (b) for low, lightweight units for easy turning in close quarters; (c) for light machines moving in small areas; (d) for small areas where machine must move sideways; (e) for heavy loads and movement over long distances

whose quality is out of proportion to the value of the load. Insufficient attention also is given in some cases to the goodwill of the operator who is expected to push the unit about, thus taking the chance that an excellent machine might not have maximum utility because of neglect of this point.

In *Fig.* 1 are shown conveyor trucks for assembling aircraft engines. The casters have straight forks with V-grooved wheels running on grouted angle-iron tracks. These tracks stop excessive vibration otherwise experienced when large, heavy assemblies are being moved over uneven floors. Floor surfaces are saved in this way and obstructions which might cause damage are eliminated by the self-cleaning inverted V-track.

Weight of load determines the minimum size of caster that may be employed. As a general rule, it is always desirable to use the largest size wheel the design of the equipment will permit because large wheels roll more easily than small ones.

Distribution of weight over the wheels should be given every possible consideration to provide maximum maneuverability and ease of operation. In this way it is possible to have the nonswivel wheels carry the greater load, effecting two advantages: Greater stability and lower mounting heights are possible by allowing the use of swivel casters smaller than the nonswivel ones. Easier steering and pushing are obtained by reducing the thrust on the swivel offset and by carrying the major load on the larger wheels.

Many heavy-duty casters are carrying loads as high as 10,000 pounds each and are now being used in installations for the handling of landing barges weighing as much as 50 tons on specially designed cradles with multiple unit casters. The refrigerating unit shown in *Fig.* 2 has a total weight of 12,500 pounds when loaded. The long span of the underframe dictated the use of center supporting casters in addition to those at each corner of the cabinet. For maneuverability in close quarters, all casters are swivel. Eight-inch resinoid wheels are employed for load rating and because their size provides ease of mobility when moved by hand. As the cabinet is required also to be transported several times daily by power truck, the large wheels provide easier travel with less wear on the wheels. The phenolic resin wheel was selected because travel is over wooden floors where a certain amount of oil, grease, and chips are present.

If the floor over which a machine is to travel is con-

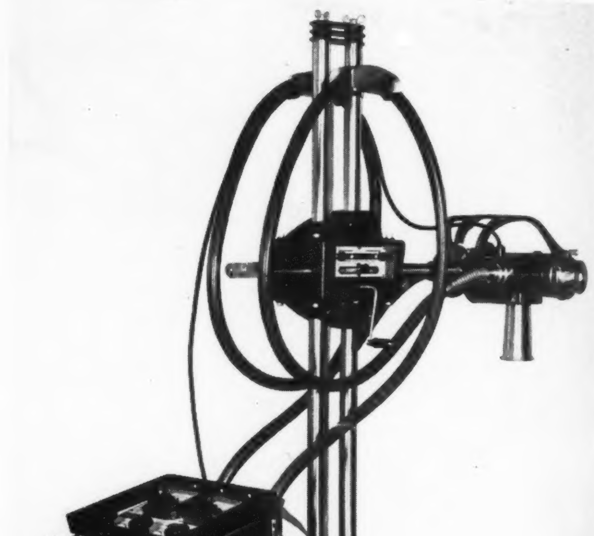


Fig. 4—Short base on six-foot high X-ray unit necessitates use of four and eight-inch diameter wheels with relatively wide faces to guard against tipping or dropping into cracks

crete, steel wheels may be employed. Often, however, the hammering effect over light bumps and the resulting wear of steel on concrete necessitates the employment of softer wheel materials or wheel tires, such as rubber or phenolic materials. These have the additional advantage of deadening sound.

For general all-purpose applications, many users have found that rubber wheels of about 80 durometer hardness are satisfactory for both the swivel and nonswivel

wheels. Others prefer to use a relatively soft material for the nonswivel wheel and harder materials for the swivel ones. The first approach, where practical, has the advantage of reduction in inventory parts, but the latter is utilized especially where it is desirable to provide shock-absorbing properties and easy operation.

A wheel rolling on a smooth floor is not subjected to crushing impacts. One that is intended to roll on a rough floor, however, may require several times the strength of the former and a larger diameter to efficiently carry the same load. For instance, an eight-inch diameter wheel will roll over a crack in a concrete floor one-inch wide without any great shock or strain but a four-inch wheel will be given a tremendous hammer blow. A crown on the wheel face is desirable, particularly for the swivel caster, to allow freedom in turning, except where the loads carried are sufficiently heavy to mar composition floors, etc.

A few typical casters are listed in the accompanying tabulation showing the type of wheel, diameter and face, together with average recommended loads based on several makes of casters. Wheel bearings for lightweight machines are usually plain, heavier loads requiring antifriction types. For the majority of applications which are under 1000 pounds, ball bearings are utilized because they take thrust and are economical for most purposes. When loads become heavier, roller or needle bearings of the precision types are employed. If thrust loads are encountered on heavier types of machines, taper bearings are utilized.

Fig. 5—Various mounting arrangements available as standard. At (a) is flat plate for mounting to base; (b) expansion stem for tubular legs; (c) stem with friction ring requiring reamed or drilled hole in tubing or base; (d) angle stem for angle-iron leg or sheet-steel body; (e) threaded stem for drilled bedplate; (f) tapered thread stem for pipe fitting; (g) large stem for welding or setscrews; (h) special bracket for edge mounting

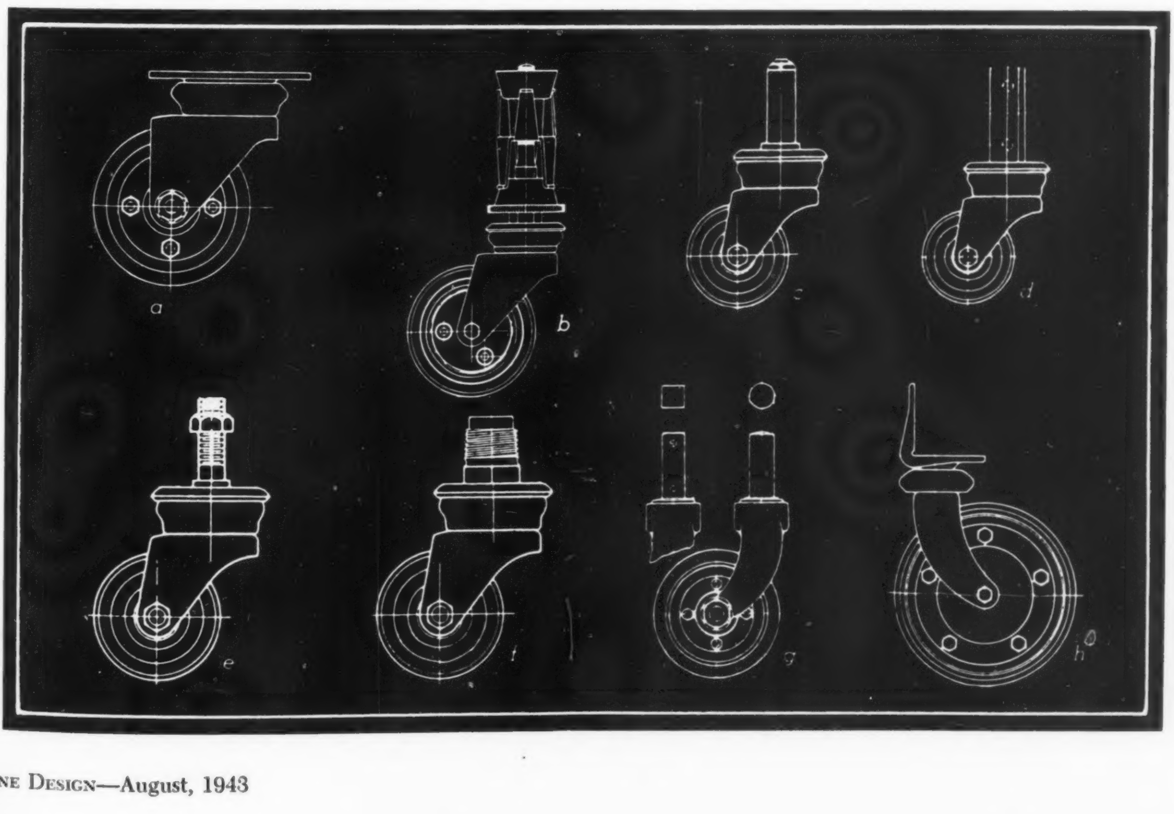
Movement desired depends on the service for which the unit is intended, whether traveling long trips, maneuvering in close quarters or combining both. Basically there are five general arrangements of wheels as shown in *Fig.* 3. At *a* is shown the most common and best suited for general-purpose mobility. It is the most practical for long distances, it steers well and has maximum stability. Pushing handle is at the swivel-caster end because from this end the rolling parts are more easily steered. If pushed from the other end, steering is difficult in that it is necessary to exert sufficient thrust against the nonswivel wheels to cause the swivel to turn.

Diamond arrangement shown in *Fig. 3b* is useful where it is desirable to pivot in machine length. It may be pushed equally well from either end and is especially adaptable to taking turns and curves easily besides handling well on straight-away. Because of the arrangement, some stability is sacrificed due to restricted bearing points.

Three-wheel style, *Fig. 3c*, with single swivel at "pushing end", has all of the advantages of *Fig. 3a* except that it is considerably unstable. This design is restricted to light loads and for movement in small areas. Most of the weight necessarily must be over the rigid forks to reduce tipping hazards.

In *Fig. 3d* is a four-swivel arrangement which is commonly used for office equipment such as adding machines. It is highly maneuverable in small areas and may be pushed in any direction within confined quarters. It is not satisfactory for long distances. For heavier, large machines this arrangement may be expanded to three or four pairs of casters as shown in *Fig.* 2. For heavy loads that are pushed long distances, the design shown in *Fig. 3e* may be used, employing a pair of central rigid fork casters and a pair of swivels at each end.

The mobile X-ray unit, *Fig.* 4, weighing approximately 500 pounds, uses eight-inch wheels at the front and 5-inch swivel casters at the back or pushing end. Wheel



face is selected as wide as possible to reduce possibility of a wheel dropping into a crack and upsetting the delicate six-foot high instrument. Due to the mobility required in small spaces the wheel base is of necessity short, increasing the stability problem and necessitating that the center of gravity be kept as low as possible.

With respect to mounting methods there is a large variety available, typical of which are those shown in *Fig.* 5, for both rigid-fork and swivel types. As with the selection of wheel diameter and face, it is advantageous to utilize standard fittings. Most generally used is the flat-plate type shown in *Fig. 5a*. Readily bolted to the base or a frame member, mounting dimensions vary with the size and load capacity required. *Fig.* 5*b* shows a common type of expansion stem caster. Designed to fit ordinary commercial tubing, it is more economical to assemble than other forms of stem casters but is more expensive. Expanding stems utilizing rubber sleeves are also employed for this purpose. Best use is for commercial tubing that is too thin to thread or where other restrictions in manufacturing are met. *Fig.* 5*c* illustrates a plain stem with a friction snap ring fitted into a groove at the upper end of the stem. This type is designed for use in small diameter tubing or pipe. Drilling or reaming is required, however, for the close tolerances needed. At *d* is a stem for fitting inside angle-iron legs or vertical surfaces of formed bases. It may be welded, riveted or bolted to the assembly. There are many variations of threaded stem, two typical ones being shown at *e* and *f*. The former is for attachment to any type of drilled base where a locking nut is wanted, whereas the latter is for standard pipe fittings. Round or square stems of the type shown at *g* are for large size casters and are attached either by welding or setscrew. Square stems are drilled for riveting or bolting if desired. Angle bracket for attachment to a box-like body is shown at *h*.

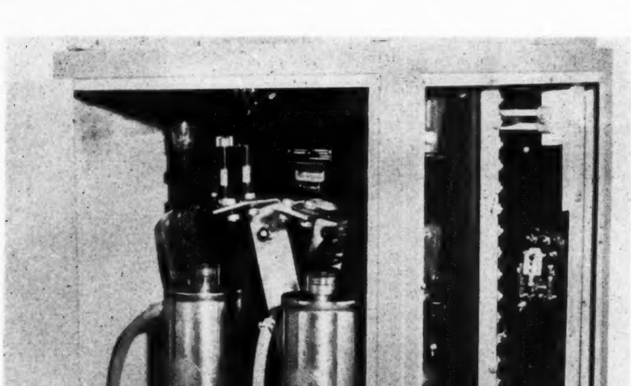


Fig. 6—Spot-welding control unit employs floating-hub swivel casters to protect delicate equipment from shocks

Typical Load Ratings

Type	Diameter (in.)	Face (in.)	Capacity (lb)
Phenolic	2	¾	75
	3	⅞	85
	4	⅞	100
	5	⅞	125
Phenolic—Heavy Duty	8	2½	1400
	10	2½	1500
Rubber Tired	3	⅞	85
	5	⅞	100
	6	2½	430
	10	2½	600
Steel	2	⅞	75
	2½	1⅛	200
	3¼	1½	350
	5	1¾	525
	6	1½	550
	8	1½	600
Steel—Heavy Duty	5	2½	1200
	6	2½	1200
	8	2½	1500
	10	2½	1600
	10	3½	10,000

Offset of swivel, often called rake or lead, is more or less standard, increase of offset requiring a special design. Offset selected as standard has the wheel center within the vertical projection of the swivel bearing, giving a swivel effect without undue thrust on the swivel bearing and kingpin. Larger offsets would require larger swivel bearings for slightly increased ease of turning. Increasing the offset also tends to decrease the stability of the unit, multiplying the hazards of tipping when the swivels are turned inward.

Swivel bearings usually are of the double ball race types because of the necessity for easy turning against the thrust exerted by the offset of the fork. Some utilize taper bearings on the kingpin stem in addition to the thrust ball bearing to effect an easy-turning fork. Correspondingly larger holes are required, however, in the machine base to provide for the increased diameter.

Brakes of various types have been employed to hold a machine in its operating position. These may be incorporated in the casters themselves or designed separately. Common types of special designs include snubbers which lower to the floor or retractable casters which set the machine on auxiliary feet.

Because of the scarcity of pneumatic tires, several methods have been developed for protecting delicate instruments in mobile equipment. One such development is a "floating hub" caster shown mounted on the spot welding unit in *Fig.* 6. In this type the load is carried off center in such a way that the wheel tends to ride over bumps without lifting the load, due to the action of a shock-absorbing spring on the eccentric hub. This mounting also tends to keep all wheels in contact with the floor and reduce distortion due to uneven floors.

The right casters properly applied will facilitate movement of mobile equipment and reduce maintenance costs. These factors are sufficiently important to warrant adequate attention to proper selection and specification.

Cooperation of the following companies in supplying illustrations and data is gratefully acknowledged: The Bassick Co., *Figs.* 3 and 6; Buffalo Caster and Wheel Corp.; The Colson Corp., Fig. 5; Divine Brothers Co.; Faultless Caster Corp., *Fig.* 1; Nice Ball Bearing Co.; Picker X-Ray Corp., *Fig.* 4; and Rapids Standard Co., *Fig.* 2.

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(lb)
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Wartime Metallurgy Conserves Strategic Materials

Part XIII—Alloying Elements

By R. E. Orton and W. F. Carter
Acme Steel Co., Chicago

HAVING discussed the general effects of the addition of alloys in the last issue, this article will consider the specific effects of each element or commonly used combinations of elements.

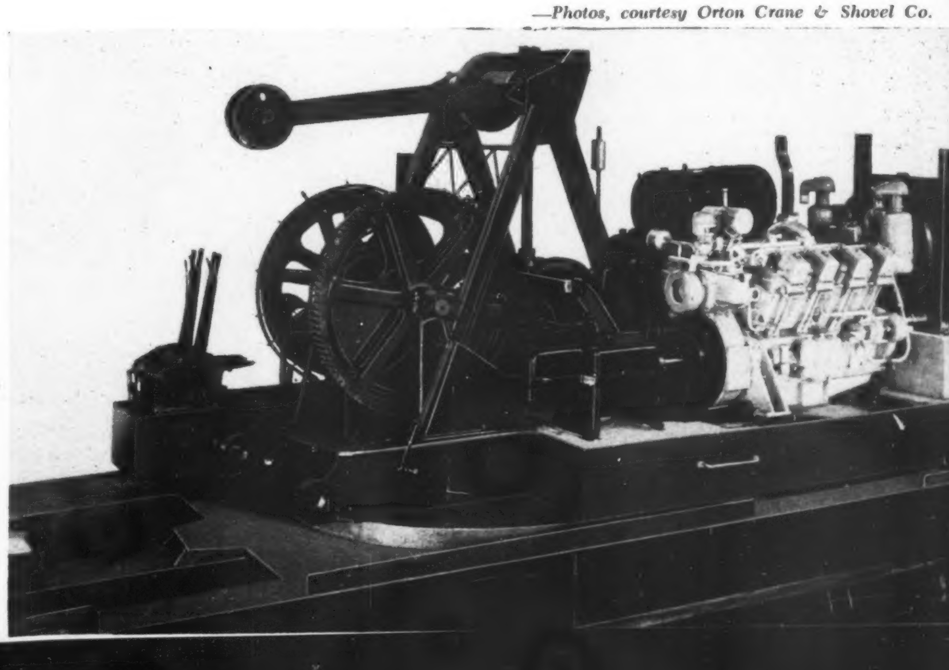
Emphasis has been placed on the importance of employing carbon and low alloy steels to secure a wider disposition of the nation's alloying elements during the war emergency. An interesting example of such an application is shown by the crane in *Fig.* 104. To handle capacity loads perpendicular to the track the "center outrigger" beam, seen projecting from under the lower structure, is extended and provided with suitable support at its end. This heavily loaded member, designed into a very small space, has a 15-inch wide by 2¾-inch thick flange. It is cast from a .65 per cent Cr, .95 Mn, .40 C steel and, after suitable annealing to refine the structure, is quenched in oil and tempered at 1100 degrees. Sufficient alloying element has been provided to assure a fine pearlitic structure of high strength and excellent ductility.

Utilization of alloys in machinery will sometimes pay in the design far beyond their intrinsic value. Through the use of heat-treated alloy shafts and gears in the operating machinery of *Fig.* 105, a reduction in the size of these members is possible permitting similar reductions in all other related parts. This allows a more satisfactory disposition of the working parts in the narrow confines of

Fig. 104—Top—Modern alloy steels have made possible the development of compact, efficient, heavy-duty locomotive cranes. Center outrigger beam and crane hook are interesting examples of structural applications of low-alloy, heat-treated steels

Fig. 105—Right—Operating machinery of crane in Fig. 104. Power shafting is SAE 4140, gears are 4145, toggle, sheave and other large size high-wearing pins are 4615 cased; oil quenched, tempered

—Photos, courtesy Orton Crane & Shovel Co.



the structure, facilitating both operation and maintenance.

Rather than making any pretense to being a handbook for the selection of steels, the intention of this series of articles is rather to present the effects of changes in composition and heat treatment on the metallographic structure and physical characteristics of the steel, leaving for the many well-recognized sources the supplying of exact quantitative information. A word of caution is in order, however, in regard to the use of the physical property charts common to such handbooks. In general they are based on small sizes (¾ to 1½ inches in diameter) and are average figures intended to be representative of commercial heats of steel commercially heat treated.

Due allowance in design work must be made for variations from the average and for larger sizes. In parts of any mass the strength at the center, after the quench, will be lower than that of the outside surface, and with large sizes the outside strength will also drop. *Figs.* 106 and 107 show hardness traverses of 5-inch diameter rounds—data being from *U.S.S. Carilloy Steels* and from Bain(1)†. *Fig.* 98 of the preceding article covered different sizes of a .40 C steel. While it is true that the tempering operation reduces this spread, generally it will still be present to some extent. Whenever considering physical properties attention must be paid to the size of section treated and tested, and to the location of the test bar in the section. Serious errors will be introduced in working with large sections if this is ignored.

The accompanying tabulation shows the more common effects of alloying elements in constructional steels. Data includes the average percentage of added elements; base warehouse price of rounds without extras; tendency to form carbides which will remain undissolved, contributing to fine grain and, in some cases, to wear qualities; relative sluggishness in dissolving in the austenite, necessitating higher or longer heating to obtain hardness after the quench; the heating critical temperature which determines the minimum quenching temperature; hardenability in terms of the size round that will harden beyond rockwell C50 for half its diameter; the cooling rate required for rockwell C50; and resistance to tempering.

Here, as in most charts, the data is average and not minimum; considerable variation can be expected from one plant to another with alterations in the heat-treating practices, changes in chemical content within specification range, grain size and so forth. Data have been extracted from the references included with this and the

†References in parentheses are listed at end of article.

The Effect of Adding Alloying Elements to Steel[1]

Added Elements and Steel Type	1935 S.A.E. Specification	Feb. 1, 1943 A.I.S.I. Specification	Per Cent of Added Elements: First	Second	Third	Base Price[2]	Carbide Forming Tendency	Relative Rate of Solution and Homogenization in Austenite	Upper Critical on Heating[3]	Size of Round That Will Harden for ½ Its Dia. to Rockwell C-50[4] and Brinnell Hardness at Center: Water Quench Dia.	Water Quench Center Hard	Oil Quench Dia.	Oil Quench Center Hard	Resistance to Tempering at 1000F[5]	Cooling Rate F Sec. 1300 Obtain Rock. C-50
.45% Carbon Steel	1045	C 1045				3.70	—	Very Rapid	1450	1⅛	225	⅝	425	—	125
Carbon Spring Steel	1095	C 1095				5.45	—	Very Rapid	1360	1½	375	¾	450	—	100
Molybdenum	—	A 4042	.25			—	Strong	Moderate	1475*	1½	375	¾	425	180	100
Chrome-Vanadium	6140	A 6145	.95	.15		9.00	Strong	Slow	1455	1¾	450	⅞	350	235	75
Manganese	T 1340 X 1335	NE 1340 C 1137	1.50 1.75			4.05 For X 1335	Moderate	Rapid	1435 1420	1⅞	350	1	425	90	60
Chromium	5140	A 5140	.80			—	Moderate	Moderate	1440	1⅞	350	1	425	140	60
Chrome Bearing Steel	52100	E 52100	1.35			11.50	Very Strong	Slow	1415	1⅞	375	1	400	170*	60
Silico-Manganese	9260	A 9260	2.00	.90		8.95	Moderate	Rapid	1500	2⅜	375	1¼	450	200	42½
Chrome-Nickel	3140	A 3140	.65	1.25		7.15	Moderate	Moderate	1415	2½	375	1⅜	375	190	40
Nickel-Molybdenum	4640	E 4640	1.80	.25		8.60	Moderate	Very Sluggish	1430	2⅝	325	1½	400	220*	35
Manganese-Moly.		NE 8442	1.45	.35		—	Strong	Sluggish	1450*	2¾	400	1⅝	460	250*	32½
Nickel	2340	A 2340	3.50			8.85	None	Slow	1360	3	375	1¾	460	80	27½
Chrome-Molybdenum	4140	A 4140	1.05	.20		6.90	Strong	Sluggish	1460	3	375	1¾	460	240	27½
Ni.-Cr.-Mo.-Mn.-Si.	—	NE 9442	.35	.30	.11, 1.15, .50	7.40	Moderate	Slow	1450*	3	410	1¾	475	180*	27½
Chrome-Nickel	3240	A 3240	1.05	1.80		8.25	Moderate	Sluggish	1425	3⅛	400	1⅞	475	200	25
Nickel-Chrome Moly.	4340	A 4340	1.80	.80	.25	8.85	Strong	Very Sluggish	1425	4¼	460	3⅛	480	250*	10
Chrome-Nickel	3250	—	1.10	1.75		8.25	Moderate	Very Sluggish	1375	5½		4½		200	5

*Estimated

(1) Conservative figures representative of commercial heats of steel, grain size 6 to 8, commercially heated and quenched.

(2) Chicago warehouse price on 500/1000 pound lots, hot-rolled, annealed rounds except 1045 and X1335 as rolled.

(3) Quenching temperatures will run 50 to 150 degrees higher, depending upon heating practice and alloys.

(4) Values will be improved by longer and higher heating, vigorous agitation of the quench, alloy content on high side, etc., reduced by opposite practices.

(5) Increase in tempering temperature to obtain same hardness values as plain carbon steel.

(6) This is lower than the critical cooling rate necessary for a fully martensitic structure.

preceding article. It has been carefully considered and is believed to be conservatively representative of good commercial practice. Surface hardnesses to be expected are from 550 to 600 brinell, except on the 1095 and 52100 on which values up to rockwell C66 may be expected.

In *Fig.* 108 are Jominy end-quench curves for most of the steels given in the chart. These data have been taken from (17), (18) and (19), and compared with other data where available. On the whole, surprisingly good agreement was found, particularly with the work of later investigators. An idea of the variations to be anticipated from one heat to another is given in (18) and (20), the effect of varying the heating temperature and holding time in (11), and curves of end-quench bars drawn back at various temperatures in (12).

The chart of *Fig.* 45[*] will permit of determining the rockwell value from the Jominy test and the cooling rate. For example, if it is known that at a given point the quenching practice will give a cooling rate of 18 degrees per second, from *Fig.* 45 this cooling rate exists at a Jominy of ¾-inch. If NE 8442 is used for this part the rockwell at the point in question, from *Fig.* 108, would be C42 as quenched, and if it were NE 9442 it would be C47.

Cooling rates at several points in rounds and the corresponding Jominy distance are given in *Fig.* 109. Approximate hardness traverses, similar to *Figs.* 103[§], 111 and 112, may be obtained from this figure as has been done for SAE 3240 in *Fig.* 110. From *Fig.* 109 the Jominy distance for the center of a 4-inch round, oil-quenched, is 1 13/16 and, from *Fig.* 108, SAE 3240 will have a rockwell of C33 at this distance. Midradius values are 1⅝ inches and C44; ¾ radius, 1⅜ and C35; and at the surface ⅞-inch and C43. In similar fashion the water-quench curve was obtained. The solid curves have been taken from (5). Lower values of the derived curves are indicative of the conservativeness of the data presented here. As previously stated, actual results will vary with grain size, alloy variations, and heating and quenching.

Effects of Alloys

The remainder of this discussion will be concerned with the specific effects of the alloying elements.

NICKEL: Nickel does not complicate the steel-making practice, practically no loss occurring in the furnace. It may be fully recovered from scrap, an important factor in the development of the NE steels. Since no carbides of nickel are formed, it is invariably present in constructional steels dissolved in the ferrite, in which it acts to strengthen the matrix of pearlitic steels and perhaps, to a slight extent, that of martensitic structures.

Presence of nickel lowers the upper critical temperature. For example, on heating SAE 2340, it is dropped nearly 100 degrees below that of a plain carbon steel, while on *slow* cooling it is well over 200 degrees (14). Theoretically, this permits of quenching from lower temperatures, thereby reducing warpage and danger of cracking. Practically, owing to the sluggishness of nickel-bearing austenite (compared with that of an unalloyed) in picking up the carbides and diffusing to a homogeneous distribution, this quality is of little commercial value. It is necessary instead to heat appreciably above the critical to assure solution and homogenization. It can be utilized only by lowering the temperature prior to the quench, a practice theoretically sound and desirable but seldom employed commercially.

[*]MACHINE DESIGN, Dec., 1942.
[§]MACHINE DESIGN, July, 1943.

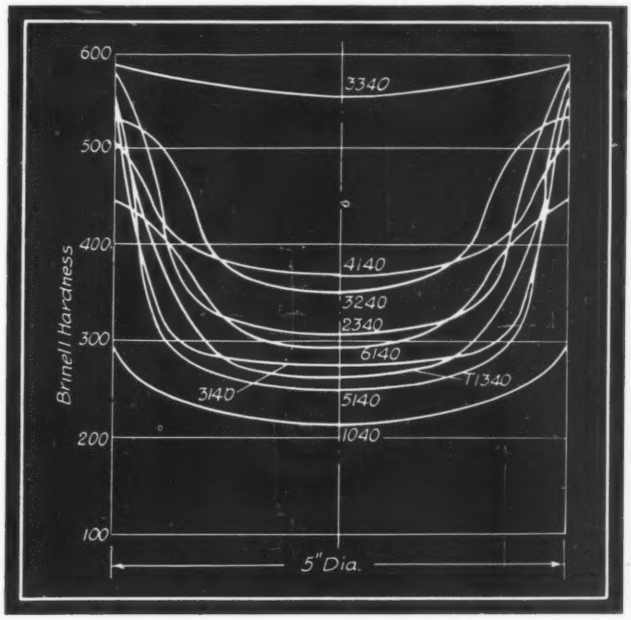
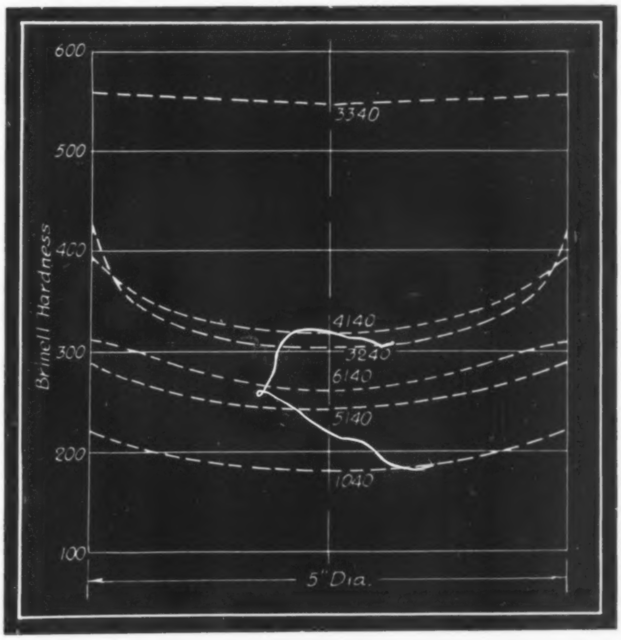


Fig. 106—Above—Hardness traverses across 5-inch rounds of various SAE alloy steels, water quenched

Fig. 107—Below—Hardness traverses for various SAE steels shown in Fig. 106 but with oil quench



Nickel in the amounts generally used contributes a marked increase in hardenability, as disclosed in the table previously discussed, and the Jominy curves of *Fig.* 108. It has some value in resisting grain growth, the effect being not nearly as great as that of the carbide-forming elements. Like most alloys it reduces also the carbon content necessary for the eutectoid. Nickel also imparts some resistance to atmospheric corrosion.

Nickel is reported as reducing slightly the rate of carbon penetration on case hardening. It has the same value

as other alloys in promoting hardenability of the case, permitting oil quenching on sections that might otherwise have to be water quenched, and increasing core strength. On large case-hardened sections, where the core structure remains pearlitic, its ferrite strengthening gives a definite improvement in core properties.

Probably the oldest alloying element added to steel, nickel remains outstanding. While unquestionably demanded for many unnecessary applications, there is no denying its value for many stringent requirements. Used in early days in "as-rolled" structural steels, its use now includes heat-treated shafting, gears and machine parts of all types; semi-tool steels where, for applications involving high stress concentrations, it seems to be almost without a substitute; and a variety of specialized uses as stainless, heat resisting, austenitic manganese as well as other alloys. Impact values of nickel steels continue well down to extremely low temperatures, particularly as used in the pearlitic form. Endurance limits are excellent, Bullens (3) reporting 2 per cent stay-bolt steel as having a ratio of 73 per cent of its ultimate, a value that exceeds the 64 per cent ratio of the yield.

From Janitsky's data (14) and that of (34), there would seem to be little justification for the common assumption that nickel steels, in the fully hardened condition and tempered to the same hardness values, have *higher* ductility or yield than other alloys. Actually, there would seem to be little difference between the various alloys, nowhere near as much variation as the variables of heat-treating practice.

CHROMIUM: Chromium offers some complication to steel making, oxidizing from the bath with appreciable losses both from scrap residues and additions. Its cheapness however, and the lower quantities employed compared with nickel, offset somewhat this loss. In the absence of carbon it will be present dissolved in the ferrite, acting to strengthen the matrix much in the fashion of nickel. It is used for this purpose in the "low-alloy, high-tensile" steels in amounts running generally from ½ to 1½ per cent, with carbon under .10 per cent. In the presence of sufficient carbon the bulk of the chromium will be present as a carbide. In the amounts common in the SAE steels—.8 in 5140 steel and 1.35 in the 52100 chrome-bearing steel—it is present chiefly as carbide.

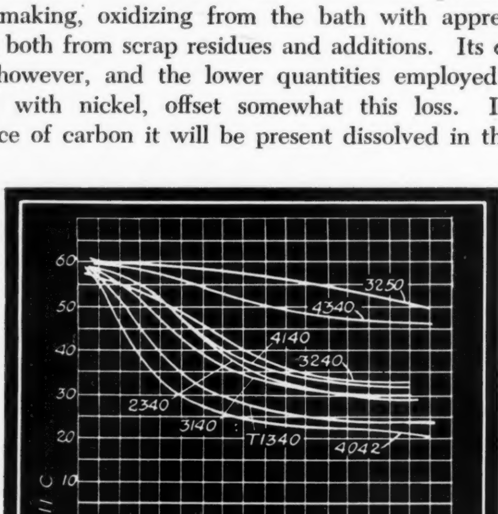


Fig. 108—Jominy end-quench tests on a number of .40 carbon alloy steels

Potential Hardenability Is Not Realized

Chromium carbides are rather sluggish in absorption into the austenite, requiring higher heating for full solution and hardenability. When not in solution they act to restrict grain growth so that generally full advantage is not taken of the potential hardenability.

Chromium introduces a mild resistance to atmospheric corrosion, increasing with amount up to 12 per cent at which a marked improvement is noted. Carbon absorption in case hardening is reported improved by this element (3), generally resulting in a higher surface carbon content. This makes chrome carburizing grades excellent for parts calling for high wear resistance.

Next to nickel in age of use, chromium has found wide application, particularly in heat-treated stocks. While employed primarily in combination with other alloys it has considerable use in the chrome-carbon grades. Chromium carbides are more wear resistant than iron carbides. It is necessary, however, that a considerable quantity of carbon be present and that the carbides be not dissolved fully. It is this quality together with economy that has resulted in the almost universal selection of SAE 52100 for ball bearings and applications such as wear plates, ball mills, crusher machinery, etc. (21).

CHROME-NICKEL: The combination of nickel and chromium gives an alloy that has for long been justly popular. The nickel will be present dissolved in the ferrite whereas in all but the very low carbon grades the chromium will be present primarily as the carbide. The one acts to strengthen any pearlite present, the other introduces slip interference in the hardened zones, and together they improve hardenability. Until a few years ago this combination unquestionably formed the most popular alloy steels for gears, shafting and other machine parts. Its occupation of this field is being challenged today by the lower price and higher hardenability of the 4100 series, and by the NE grades.

Compared with the straight nickel series, the 3100 steels show improved machinability; compared with the chrome series they show improved hardenability. Probably the only exceptions that should be made in the preference of this combination to either alloy would be for certain special applications. For instance, extremely low temperatures and boiler and stay-bolts etc., require the nickel types and ball bearings and other parts calling for high abrasion resistance utilize chrome-bearing steels.

In considering an alloy for large sections, ferrite strengtheners deserve emphasis. It is impossible to obtain a martensitic structure at the interior of very large forgings and shafts, but only emulsified pearlite. In such case the value of elements dissolved in and strengthening the ferrite becomes pronounced. For this reason the chrome-nickel series is a good base for large sections, the carbide-forming element serving as a cheap hardening

agent, the nickel improving this hardenability and strengthening the ferrite.

MOLYBDENUM: Molybdenum is recoverable from scrap and offers little complication to the steel-making process. In low-carbon steels it acts as a ferrite strengthener, the effect for equal quantities being about the same as that of chromium, somewhat more than nickel. One per cent in a normalized steel will add some 10 to 12,000 pounds per square inch to the ultimate, 8 to 10,000 to the yield. The element is strongly carbide forming, and if any appreciable amount of carbon is present, little of it will be present in the ferrite. Its carbides are sluggish in dissolving and diffusing in the austenite, so that the hardenability will vary widely with the heating time, temperature and the microstructure previous to heating. These same carbides, of course, contribute greatly to grain-growth resistance.

As usual, the sluggishness in dissolving in the austenite is accompanied by sluggish tempering characteristics and good creep resistance. Adding one-half per cent moly to a low-carbon grade doubles the creep resistance at temperatures from 800 to 1000 degrees. The carbon-moly steels are widely used for valve castings for high temperature steam, refinery service, etc.

Molybdenum is not commonly used by itself in constructional alloy steels. Grade A4042, which should perhaps be classed as a manganese-moly, was employed to some extent in the beginning of the present emergency, but shortages of the element and need to conserve it for other uses resulted in the discontinuance of the general manufacture of this grade. Chief use of moly is in combination with other elements.

Molybdenum-bearing steels show appreciably improved machinability at the same hardness, particularly in the heat-treated condition (16).

Are Best for Durability

NICKEL-MOLYBDENUM STEELS: The nickel-moly through-hardening steels are not widely employed, although a good many of the arguments for the chrome-nickel steels would apply to this combination. This is probably accounted for by the appreciably higher cost of 4640 compared with 3140, with but little improvement in hardenability. The combination is, however, extensively employed in carburizing grades, 4615 with 1.85 Ni and .25 Mo being highly favored for parts demanding the best in wear and durability such as aircraft gears, truck differential gears, etc. The nickel serves as a ferrite strengthener, imparting high ductility and hardenability to the core, and the molybdenum carbides improve the abrasion resistance and enhance the hardenability. It is also reported as affording lower warpage and residual stresses than other steels of equivalent hardenability, the last being enhanced by its good resistance to the tempering operation.

CHROMIUM-MOLYBDENUM STEELS: This combination is a relatively new introduction to the family of constructional alloy steels. It has been developed to a high state of perfection, primarily by the aircraft industries. Because of its excellent combination of hardenability, general reliability in heat treating, machinability and low cost, it is today supplanting many other complex steels.

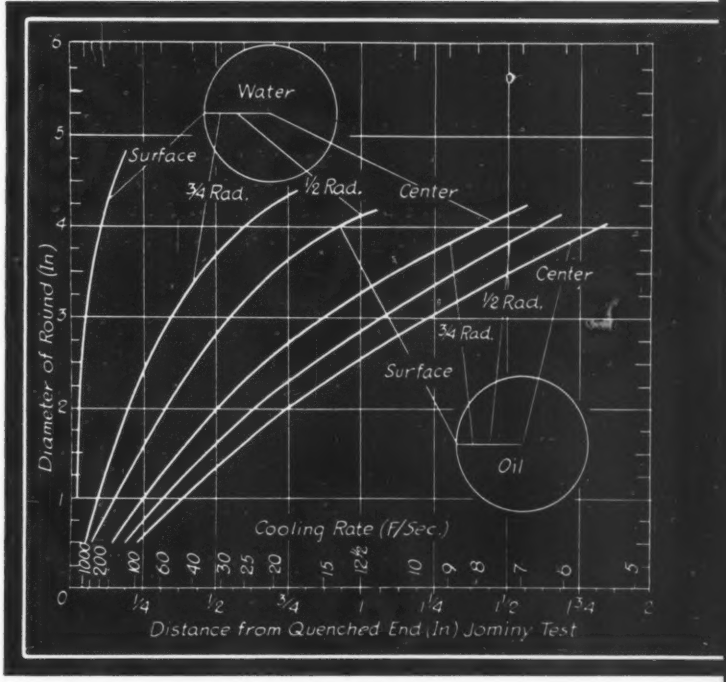


Fig. 109—Jominy distances that have same cooling rate as various sizes of round bars for oil and water quenches

The 4130 grade is rolled into strip and drawn into tubular shapes and as such is widely used for aircraft fuselage construction. It has good formability and machinability as well as sufficient alloys to make a welded joint in the thicknesses common in this work, air hardening mildly. Generally used in sections under ¼-inch thick and in the normalized condition, 90,000 ultimate and 70,000 yield with good ductility may be anticipated.

Reaching wide acceptance for many machine parts of medium section is SAE 4140. Depth-hardening values are better than 3140 and equal to 2340 or 3240, with lower cost and machinability than either. Warpage is generally reported as less, making it desirable for parts not machined after heat treatment.

High chrome-moly steels, in particular with 5 per cent chrome, and one-half per cent moly, are widely used for high temperature petroleum service. The moly imparts good creep resistance and the chrome resists corrosion.

NICKEL-CHROME-MOLY: A grade of high alloy that has found wide acceptance for large forgings and shafts, or unavoidably high stress concentrations, is nickel-chrome-moly, of which SAE 4340 is representative. This combination has the highest hardenability of any listed in the table except 3250. It is highly resistant to tempering, permitting of good stress relief (22); accompanying this is the usual reluctance in entering into solution in the austenite. Of outstanding importance is its remarkable machinability at high hardness, the production of heavy duty truck axles at 400 to 444 brinell with 230,000 ultimate, 210,000 yield being not uncommon, and 500 brinell having been satisfactorily machined (16). These steels are air hardening in light sections, and for high-production machining must always be thoroughly annealed. Such heat-producing operations as the use of a cut-off wheel will produce a fin and surface of high hardness which may result in cutter failure.

The nickel-chrome and the high manganese steels are

subject to a grain boundary precipitation phenomena known as "temper brittleness". Steels of this class, slowly cooled from draw temperatures ranging from 800 to 1200 degrees, may show a marked reduction in impact. While relief may be obtained by quenching, this is not always permissible nor is it possible in large sections. It has been shown that the addition of one-fourth to one-half per cent moly will obviate this effect (3), being a definite advantage of this class of steels over the straight nickel-chrome grades. Even the 3½ per cent nickel grades at times show this effect.

A popular grade of "semitool" steel has .70 C, .60 Cr, 1.75 Ni, and .20 Mo. If held sufficiently at temperature to give good solution it will be found air hardening in sections up to about ½-inch, with low distortion. Its nondeforming properties are also as good as oil quenched, and the high carbide gives good abrasion resistance. It is reported as giving on a 1-inch section, oil quenched and drawn at 400 degrees, 300,000 ultimate, with 280,000 yield, 5 per cent elongation in 2 inches and 12 per cent reduction in area. It has been used with marked

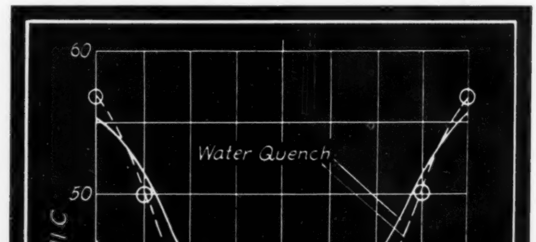


Fig. 110—Calculated hardness traverses of SAE 3240, shown dotted, compared with actual traverse, solid lines

success for punch and die designs carrying unavoidably high stresses, although of course with some loss in wear over conventional die steels. It has also been used for shears, cams, gears, collets, etc.

Vanadium: While vanadium has been employed to some extent in nearly all constructional alloy steels its use alone is not common, there being no provision for it in the SAE series. It is strongly carbide forming, is sluggish in dissolving in the austenite, and correspondingly improves resistance to tempering. The chief advantage in its use is grain-size control, scavenging and deoxidizing of the melt, all of which are supplied today by aluminum additions. In the amount usually employed it contributes little to the hardenability, is easily oxidized and is high in cost, making the application of vanadium steels seldom economical. However, Strauss and Norris (14) consider straight vanadium of value for large forgings such as locomotive axles, etc., in the normalized condition. They also recommend the addition of vanadium to large castings to promote a fine dendritic structure as well as a smaller austenitic grain.

Chrome-Vanadium: Because of its excellent grain-size control and resistance to tempering, the chrome-vanadium steels at one time formed an extremely popular alloy for both case and through-hardening where toughness was paramount. Hardenability, however, is not high, being but little improved over the carbon-moly type. It has been applied to auto and truck transmission gears and shafts, axles and differential gears, etc., although its use for such purposes is uncommon today.

In SAE 6150 or ASTM-A232-41, this alloy has been widely applied to heat-treated springs for severe service, particularly for elevated-temperature duty such as is encountered in valve springs. Zimmerli's data (23), however, does not substantiate this viewpoint. Wahl gives a further discussion of spring steels in (24).

Produces Finely Divided Nitride

Innumerable combinations of vanadium with other alloys have been employed, particularly in castings and forgings of heavy sections, and for other special requirements. Low-carbon, chrome-vanadium, with and without moly, have been satisfactorily used for nitriding. It is reported that the vanadium tends to produce a finer divided nitride deposition with consequent improved case properties.

Manganese: Manganese is a moderate carbide former, dividing between the ferrite and carbide, so that it may be employed in both roles. It makes an excellent low-cost alloying addition, and has been widely applied for both "as-rolled" structural and heat-treated steels. Quite similar in properties and about twice as potent for the same weight, it is frequently employed as a substitute for nickel. Typical type is NE 1300 series, formerly SAE T1300, (33).

Manganese is frequently added in small amounts to plain carbon, case-hardening steels, as in X-1020. While the addition is small it improves the rate of casing and hardenability sufficiently to reduce the danger of soft spots. Since it carries no price premium over 1020, and improves somewhat the machinability, it is recommended where the sulphurized steels cannot be permitted.

Manganese is used in rail steels with .55 to .70 C and about 1½ per cent Mn, with material improvement in the wear qualities. Many other combinations of alloys with manganese have been developed, too numerous to be covered in this discussion. A few of these are a Mn-Mo carburizing steel available in bar stock; a chrome-manganese .40 C through-hardening steel, widely employed in castings; a .50 C Mn-Cr-Mo; and many others.

Of considerable importance for high wear resistance are the austenitic manganese steels, running around 12 per cent Mn, 1 to 1.4 per cent C. The steel is austenitic even on air cooling. Heat treatment consists in heating to 1800 to 1900 degrees to dissolve and diffuse the elements, and quenching in water to prevent precipitation of embrittling carbides. The high wear resistance is due to cold working, generally imparted in the first service,

(*Continued on Page* 224)

Tachometers Guide Precise Operation

By B. R. Hill
Westinghouse Electric & Mfg. Co.

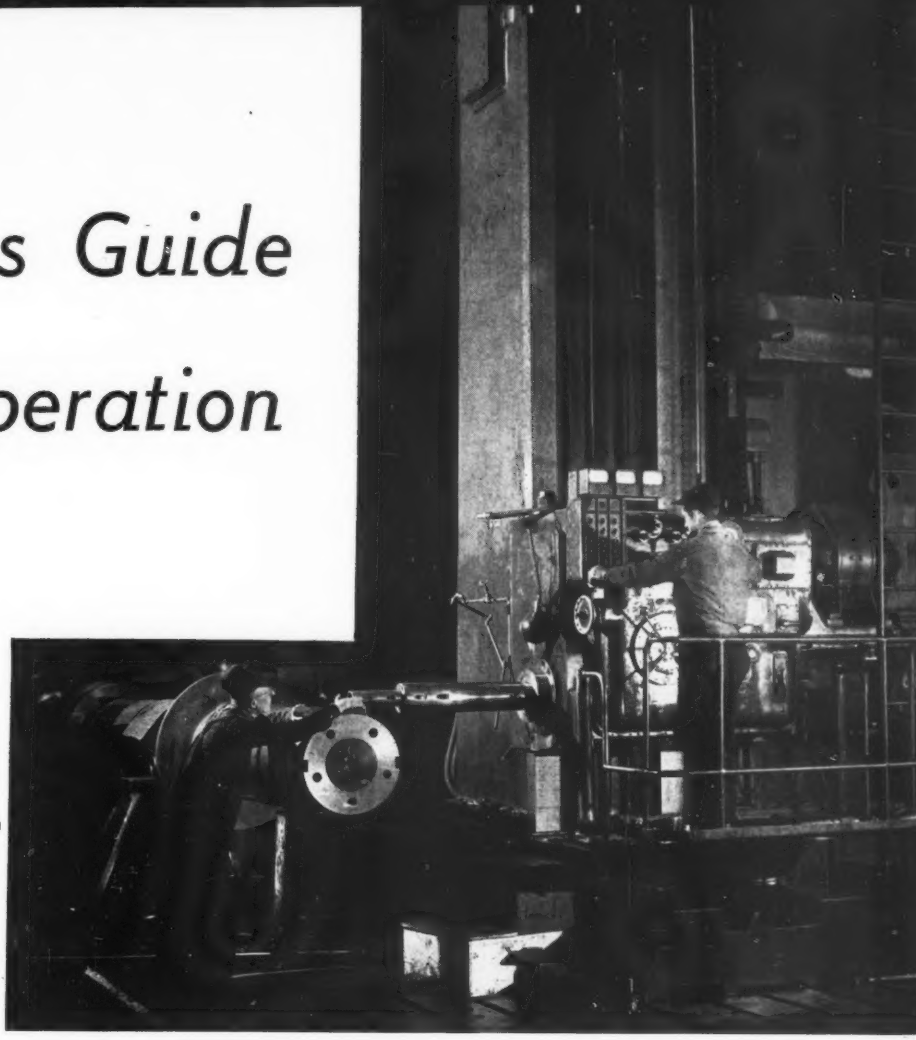


Fig. 1—Horizontal boring, drilling and milling machine has tachometer instruments mounted on head to indicate feeds and speeds

FOR many years small electric generators with permanent magnet fields have been built for the specific purpose of producing voltage proportional to generator speed. With such generators a voltmeter can have a scale laid out in terms of revolutions per minute, gallons per hour, inches per minute or tons per hour. At once an almost unlimited field of application opens up involving control of industrial processes and operation of many types of machines, typical of which is the horizontal boring, drilling and milling machine in *Figs.* 1 and 2.

Recent development of electric tachometers of this type has further extended the field of application and greatly improved the reliability as an accurate speed-measuring device. The most important improvement concerns the use of an alternating-current generator with a permanent magnet field. By means of this innovation, items requiring adjustment and maintenance such as delicate brushes and alloy commutators on direct-current generators have been eliminated. No slip rings or brushes are needed since the field-producing units and coils are stationary. The only rotating member is a laminated steel rotor which has ten slots or poles, flux changes being produced in the stationary coils when it rotates. The principle of operation is shown in *Fig.* 3. Output voltage is 10 volts per 1000 revolutions per minute and the frequency is 60 cycles at 360 revolutions per minute. Since frequency is proportional to speed, the generator is designed for the relatively high frequency to prevent oscillation of the instrument pointer at low speeds.

Frequently the shaft of the generator can be driven direct from certain parts of a machine, thereby eliminating need for step-up or step-down speed changers. Coupling can be accomplished direct or by belts, gearing, flexible shafts, or universal

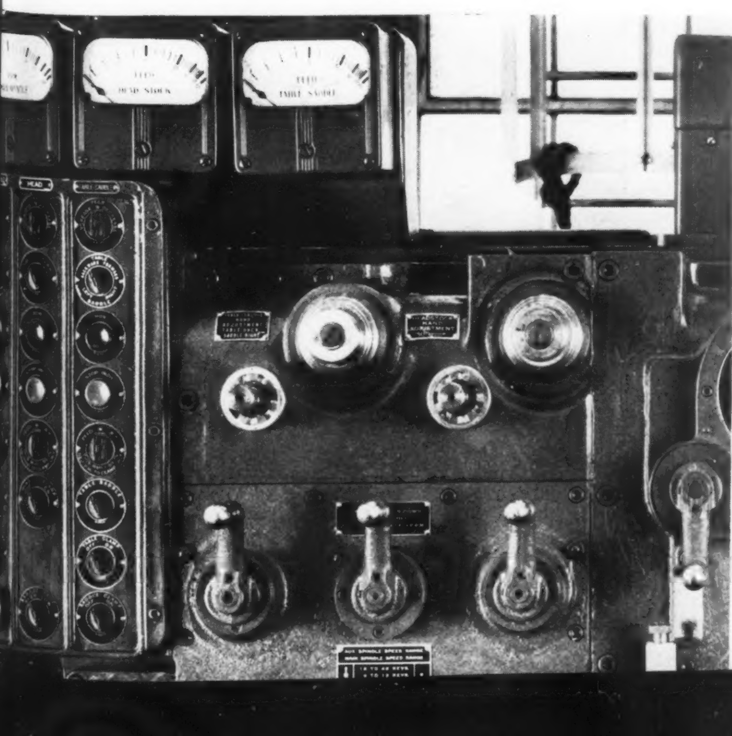


Fig. 2—Close-up of control station shown in Fig. 1. Instruments showing spindle speed, headstock feed and table-saddle feed are each connected to a generator driven at speeds proportional to final drive

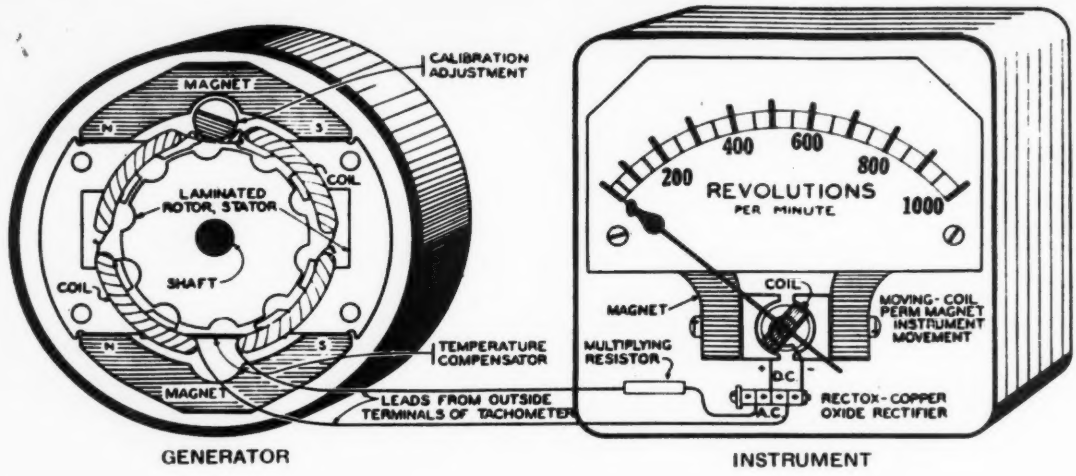


Fig. 3 — Design principle of alternating-current tachometer

joints whichever fits the conditions of the particular application best.

A tachometer generator must have certain performance characteristics essential to a reliable and accurate speed-measuring system. With no load on the generator, the voltage output plotted against speed should result in a straight line. Usually this characteristic deviates somewhat from a straight line due to the indicating instrument load. Proper instrument and generator design, however, make it possible to hold very close to the straight-line relationship. Typical output characteristics are shown in *Fig.* 4. Single-instrument loads are of the order of 400 ohms per volt; if 1000 revolutions per minute of the generator deflects the instrument to full scale, the instrument would have a total resistance of approximately 4000 ohms.

The curves in *Fig.* 4 show only a slight deviation from straight-line output. This does not give erroneous readings since the instrument dial is calibrated in terms of actual speed of the generator. The alternating-current tachometer may closely approach the uniform-scale characteristics of the direct-current tachometer and generator output is little affected by temperature change. Such temperature errors have been reduced to a minimum through use of a special alloy shunt installed in the flux path of the permanent magnet. Over a temperature range of —50 to +50 degrees Cent. the temperature error is less than .2 per cent per 10 degrees.

Indicating Instrument Rectifies Current

Voltage output of the generator is measured by a rectifier type alternating-current indicating instrument as shown in *Fig.* 3. A small copper-oxide full-wave rectifier is mounted inside the instrument case and its direct-current is applied to the usual permanent-magnet, moving-coil type of direct-current instrument. The development of the rectifier-type instrument has made possible the use of this type of alternating-current tachometer generator.

Special scale-distribution characteristics may be obtained through use of nonlinear air gaps installed at the pole pieces of the instrument permanent magnets. Some of these special applications will be described later in connection with machine tool measurements.

The copper-oxide rectifier is little affected by temperature or by frequency variations of the generator. Actually the latter is unimportant since for any given speed the frequency is constant and the instrument calibration completely eliminates frequency errors. Low energy consumption of the indicating instrument permits operation of several instruments from one generator, a distinct advantage where speed indications are desired at several locations. Then too, low-energy consumption minimizes errors due to resistance of leads between generator and instrument; in other words, long leads have a smaller adverse effect on the accuracy of the system, reducing the errors of the readings.

Various types of large precision boring, drilling and milling machines have utilized this type of tachometer successfully. Usually, the operations of such machines are controlled from a pushbutton and selector switch panel mounted on the machine or at the end of a long cable. Gear-shift and motor-field controls provide a wide range of speed and feed rates, and this type of machine therefore is often equipped with tachometers so that the operator knows what speeds are being used, *Fig.* 5. In

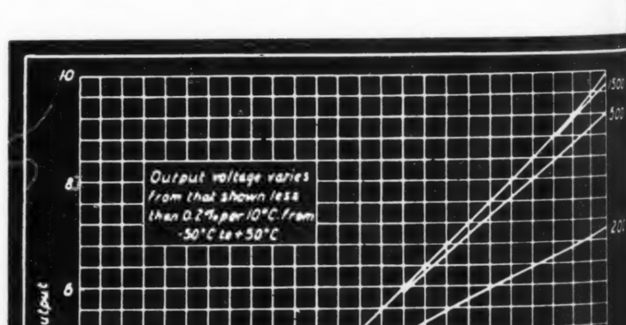


Fig. 4—Characteristic curves of generator output

performing certain types of work this information is essential for efficient operation. This is especially true for modern high-speed cutting tools and the great variety of metals and parts that have to be machined these days.

In applying tachometer equipment to this type of machine the generator must be driven by a rotating member of the machine the speed of which represents the speed or feed rate at the work, regardless of the gear ratio or motor speed being used. Where the speed ranges are extreme, it is desirable to use a multirange instrument to get readable deflections under all conditions. In this case the generator drive may be taken from a shaft ahead of the gear shift, and the generator thus will rotate at the same full-scale speed regardless of the gear ratio being used. Then each scale range on the instrument dial represents a speed or feed rate for each gear ratio. Readings are taken on the scale that corresponds to the gear ratio being used. The instrument scales are identified by suitable titles on the dial; in some cases where several scale arcs are used, colored divisions and figures may be utilized to help distinguish between ranges. With moderate speeds it is usually more desirable to have a single scale on the instrument because the single-range instrument is easier to read and is less confusing to the operator. Then the generator is driven from a shaft whose speed represents the final drive of the spindle or the feed device.

In most applications of tachometer equipment it is important that instruments be readable at the lower end of the scales. Therefore this part of the scale is expanded as mentioned previously, the permanent-magnet air gaps being shaped so that the torque increases, per unit of voltage change, as the pointer approaches zero. This gives large deflections for low speeds and a scale distribution which is tailor-made for the application. It is especially valuable on machines where the generator is driven at a speed proportional to the final drive; in such a case the single-scale arc must cover the entire range of the machine.

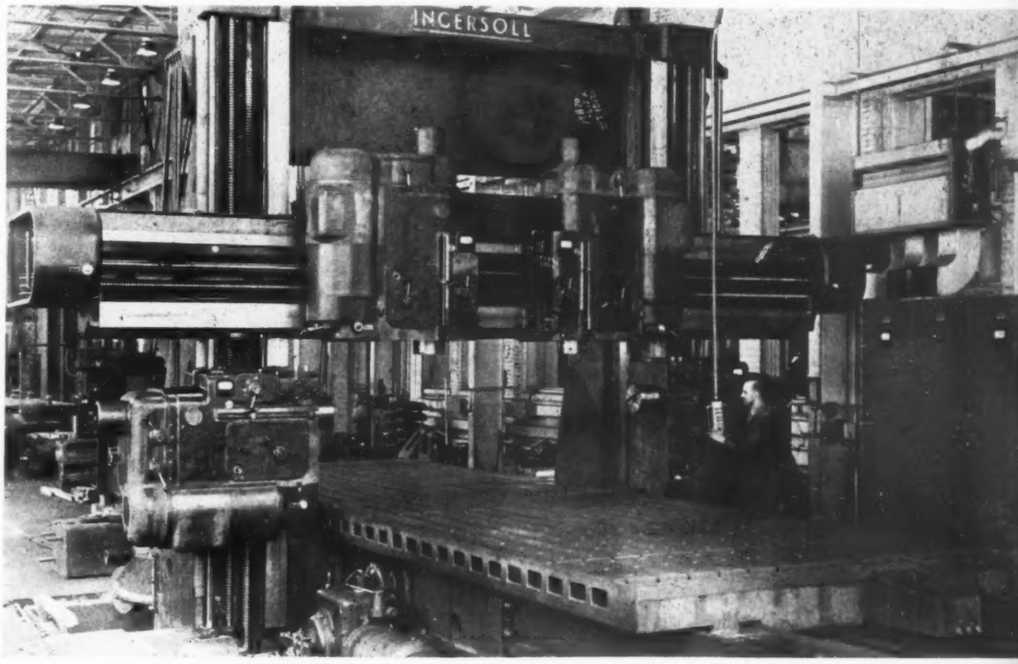


Fig. 5—Right—Milling and boring machine equipped with tachometers. An instrument for spindle speed is mounted on each of four spindles. Feed-rate instruments are mounted on control cabinet at right

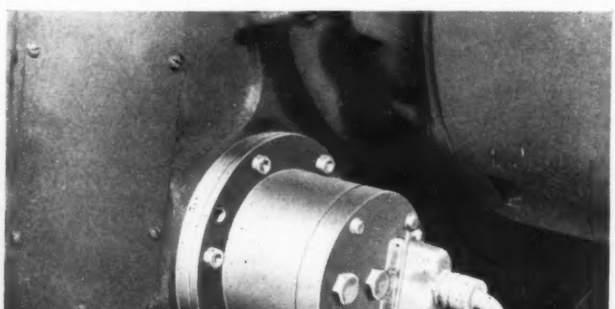
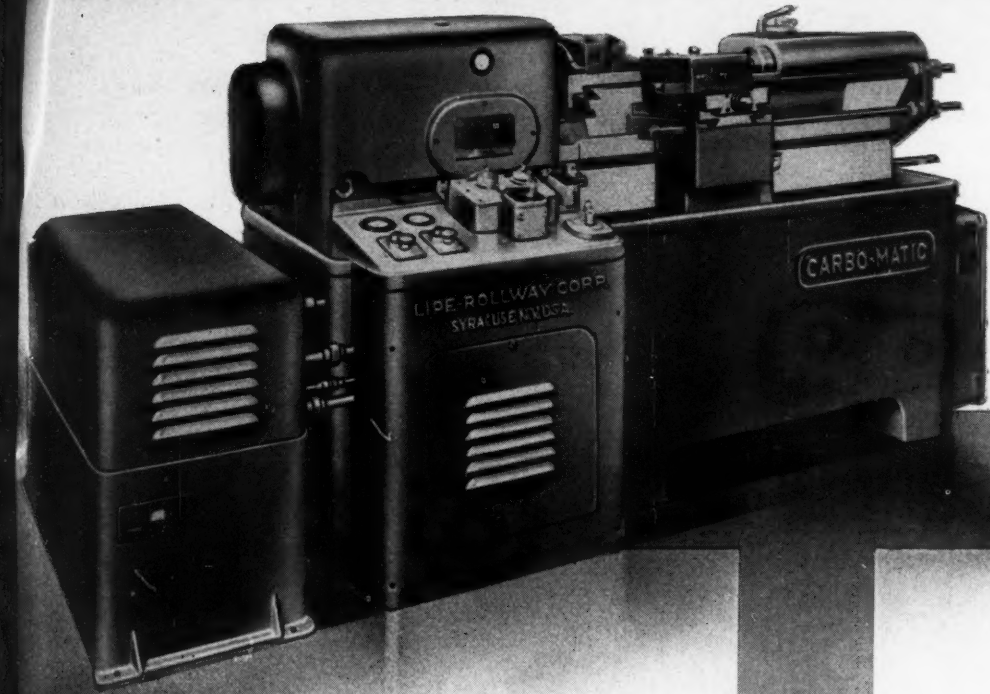


Fig. 6—Below—Tachometer generator on gearbox

The most common generator drive consists of a direct coupling to a shaft in the gearbox. *Fig.* 6 shows how the generator, mounted on a flange, is bolted to the side of the unit. Since the gearbox is filled with oil, the flange is an oiltight fit. The generator bearing is equipped with a seal to prevent entry of oil from the gearbox; the seal also prevents generator lubricant from escaping.

Encouraging!

THIS COUNTRY is practically "over the hump" of its rubber dilemma—technologically and from the sole standpoint of military needs. Research on synthetic rubber has progressed to the point that it can be converted into almost any needed war product if no more natural rubber becomes available. Several new plants are complete and ready for production; they were financed by war-emergency funds.



Left—Automatic throughout the entire cycle, the Lipe carbo-matic lathe has hydraulic control of holding equipment, feeds and tailstock quill. Power is transmitted through a multiple V-belt and cone gear drive, while the totally enclosed base and bed are of heavy construction, permitting high-speed operation with heavy cuts

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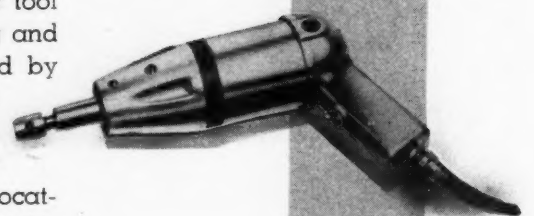
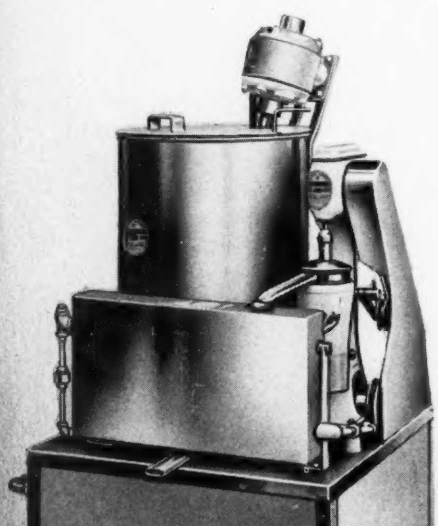
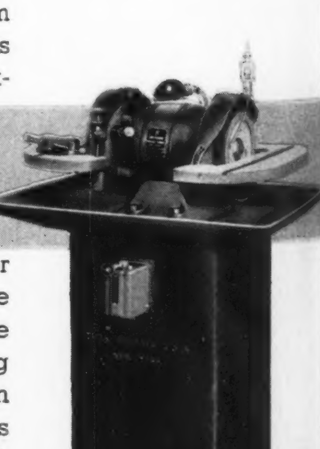


Right—Push and pull action in the H & H portable electric tool for filing, scraping, polishing and similar operations is effected by an eccentric mounted on a bevel gear, the driving motor and bevel pinion being on the centerline of the reciprocating tool holder. Pistol grip includes conveniently located control switch for starting



Left—Incorporating a heater for pasteurizing and an aircooled compressor for refrigerating, the portable self-contained Mechanical Cow produces 40 quarts per hour of milk, cream or ice cream mix from skim milk powder, sweet butter and water. Mixture is agitated during pasteurization in stainless steel mixing tank, emulsified in seamless steel tinned tubular centrifuge and cooled to 45 degrees in heavily tinned copper cooler



Right—Quick-acting indexing table on the Prosser carbide tool grinder permits instant setting to the desired angle. Reversible motor drives the spindle direct and a brake is provided for quick stopping when reversing the direction of rotation. Location of the silicon carbide cup wheels on the spindle is easily adjustable to compensate for wear

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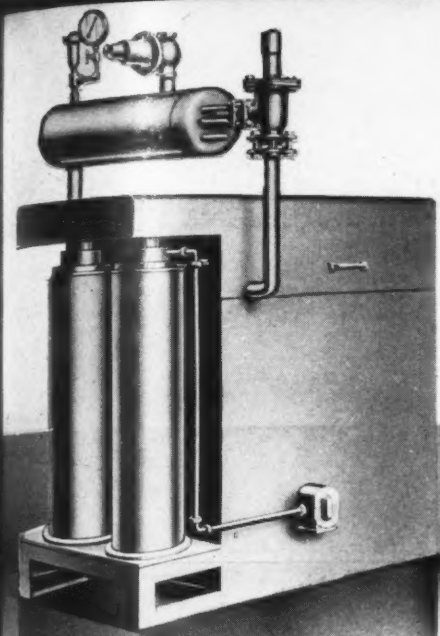

Left—Using direct-expansion ammonia as refrigerant, the Girdler votator chills liquid eggs from 50 to 31 degrees in approximately 15 seconds, inhibiting bacterial growth between breaking and packaging for shipment to drying plants. Two-cylinder unit shown handles from 3000 to 4000 pounds of eggs per hour

Above—Cam-controlled [illegible] on the Wickman Swiss-type [illegible] hardened steel with micrometer adjustment both radially and axially. Nitrided alloy steel spindle is supported on an adjustable bronze bearing and a preloaded double-row ball bearing which also carries the thrust

Left—Designed for continuous welding on a production basis the P & H alternating current welder has extra cooling capacity, simplified drip-proof construction, full view current calibration panel and specially styled canopy. Improved design of controls permits accurate setting without creeping

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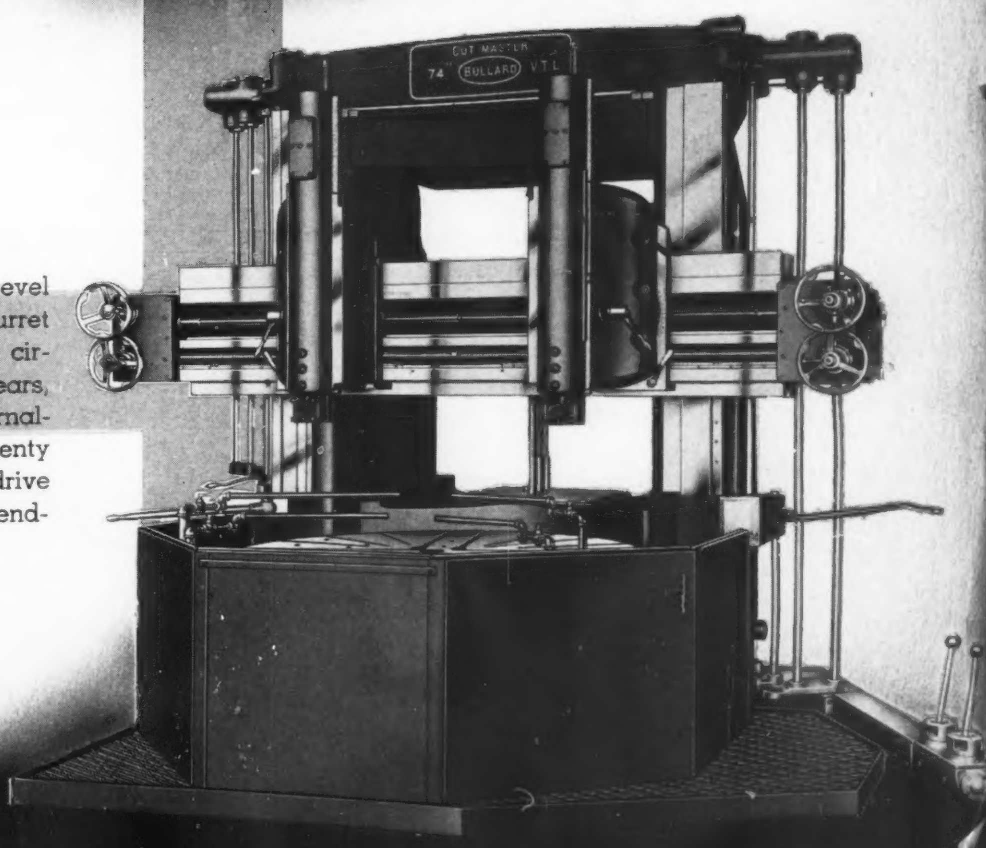


Right—Spindle bearings and spiral bevel drive pinion of the Bullard vertical turret lathe run submerged in filtered oil circulated under pressure. Headstock gears, which are constant-mesh with internal-external tooth clutches, provide twenty table speeds. Nonreversing worm drive for side head holds it in position independently of counterweight

Looking Ahead to Postwar

ENGINEERS responsible for design still appreciate to the full that their labors of the past few years have not yet borne the fruits of victory and that the final conclusion of the war may be a long way off. Yet there are distinct signs that the thinking of the profession is no longer solely occupied with development of armaments and allied equipment but rather that some attention is now being given to postwar designs.

It is fortunate for the successful prosecution of the war that such activity has not, except in relatively few instances, reached the stage of actual development of new machines. Some of the larger companies, only, have thus far been able to take advantage of diminishing activity in their engineering departments to free certain members of their design staffs intermittently to engage in postwar development. Other companies, in the medium-sized category, have reached only the research stage, and smaller concerns are just beginning to look ahead.

All in all it can be said—based on a survey of machinery builders representative of the entire field—that at least 80 per cent of such companies, regardless of size, are either planning or thinking of plans for the postwar period. In some cases active committees meet regularly to formulate such plans; in others only small groups or single individuals are assigned to the job. The amount of time and effort is bound to vary in relation with the anticipated position of each type of company after the war.

Such thinking ahead is to be strongly commended—without losing sight, however, of Donald Nelson's recent warning that "the war is not yet in the bag," or of the fact that nothing would please Hitler and Hirohito more than that we should begin to act as though our enemies were nearing a state of complete collapse. Only as the day draws nearer on which our boys will return can intensive efforts be conscientiously and increasingly devoted to actual design work for the reconversion and re-employment period that ultimately lies ahead.

L. E. Jermy

Velocity and Acceleration of Geneva Mechanisms

By J. Harland Billings
Drexel Institute of Technology

BECAUSE of its simplicity the geneva mechanism, *Fig.* 1, is popular for intermittent drives as well as for counting or stop mechanisms. Any number of slots from four up to fairly large numbers can be used, particularly if the slots and the locking surfaces are placed on separate disks. The number of turns made by the driver B for one turn of the driven member C is equal to N/n, where N is the number of slots in the driven and n is the number of driving pins in the driver.

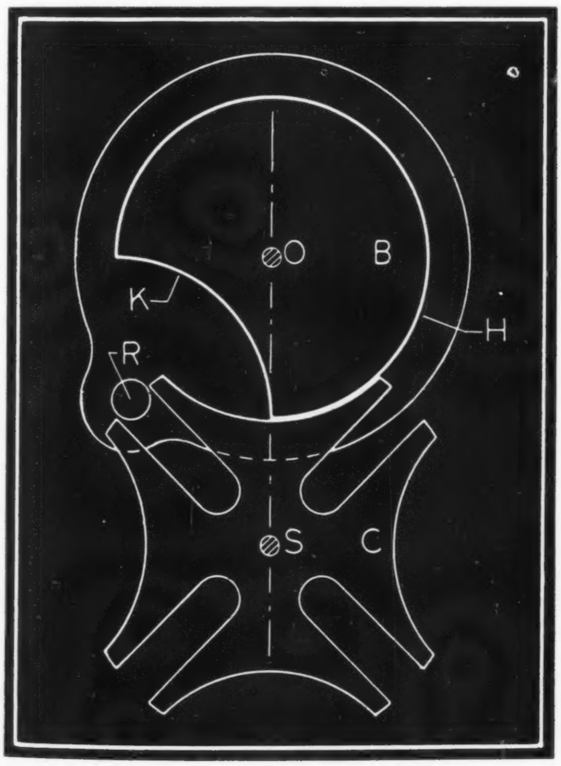


Fig. 1—Elements of the geneva stop mechanism. Angle of rotation of driving member B is measured from this position

Determination of velocities and accelerations in this mechanism is somewhat involved, and as an aid to the designer the curves in *Fig.* 2 are presented. The abscissa represents degrees of angular travel of the driving member B measured from its position in *Fig.* 1 when pin R is just entering the slot. Broken-line curves give angular velocity ratio (ω_C/ω_B) where ω_C and ω_B denote the angular velocities of the driven and driving members respectively. Full-line curves give angular acceleration ratio (α_C/ω_B^2) where α_C denotes the angular acceleration of the driven member, radians per second per second, ω_B being assumed constant.

For satisfactory operation at medium or high speeds the pin R must enter and leave the slot tangentially. This condition is evidently fulfilled if the angle OSR is made equal to π/N. If the radius to the center of the pin is $OR=r$ and the center distance $OS=c$, the condition is that $r/c=\sin\pi/N$. The curves in *Fig.* 2 apply only to mechanisms with these proportions.

Problem Illustrates Use of Curves

EXAMPLE: It is required to find the velocity and acceleration of the driven member C of a six-slot mechanism, the driving member B revolving at 200 revolutions per minute and being in the phase 40 degrees from initial contact with the slot.

The angular velocity corresponding to 200 revolutions per minute is $200\times 2\pi/60=20.94$ radians per second. From the velocity curve for six slots the value at 40 degrees is .704 and the required velocity is therefore $.704\times 20.94=14.74$ radians per second. From the acceleration curve for six slots the value at 40 degrees is 1.25. The required acceleration is therefore $1.25\times 20.94^2=548$ radians per second per second.

All curves are plotted for the first half of the travel of the follower, that is, until the pin R (*Fig.* 1) reaches the center line OS. Beyond this phase the values are repeated in reverse order, all acceleration values being negative. For the foregoing example the pin is on the center line at 60 degrees, therefore at 80 degrees the velocity would again be 14.74 radians per second

while the acceleration would be —548 radians per second per second.

For mechanisms having numbers of slots other than those included in *Fig.* 2, graphical or analytical methods may be used. The graphical method used in the construction of these curves is described in Reference 1, while a general method of acceleration analysis is discussed in Reference 2.

Results of mathematical analysis may be expressed in the form of equations as follows:

$$\frac{\omega_C}{\omega_B}=\frac{\cos(\beta-\theta)-\dfrac{r}{c}}{\dfrac{c}{r}+\dfrac{r}{c}-2\cos(\beta-\theta)} \quad \text{.......................(1)}$$

$$\frac{\alpha_C}{\omega_B^2}=\frac{\left(\dfrac{c}{r}-\dfrac{r}{c}\right)\sin(\beta-\theta)}{\left[\dfrac{c}{r}+\dfrac{r}{c}-2\cos(\beta-\theta)\right]^2} \quad \text{..................(2)}$$

where β=angle $ROS=\cos^{-1}(OR/OS) = \cos^{-1}(r/c)$, $r/c=\sin(\pi/N)$, and θ=angular travel of driving member B measured from its position in *Fig.* 1 when pin R is just entering the slot.

Often it is important to know the maximum instantaneous values of velocity and acceleration of the driven member. For the velocity, inspection of *Fig.* 1 or substitution in Equation 1 shows that

$$\left(\frac{\omega_C}{\omega_B}\right)_{max}=\frac{r}{c-r} \quad \text{.............................(3)}$$

Differentiating Equation 2 and equating to zero gives the value of θ for which the acceleration is maximum:

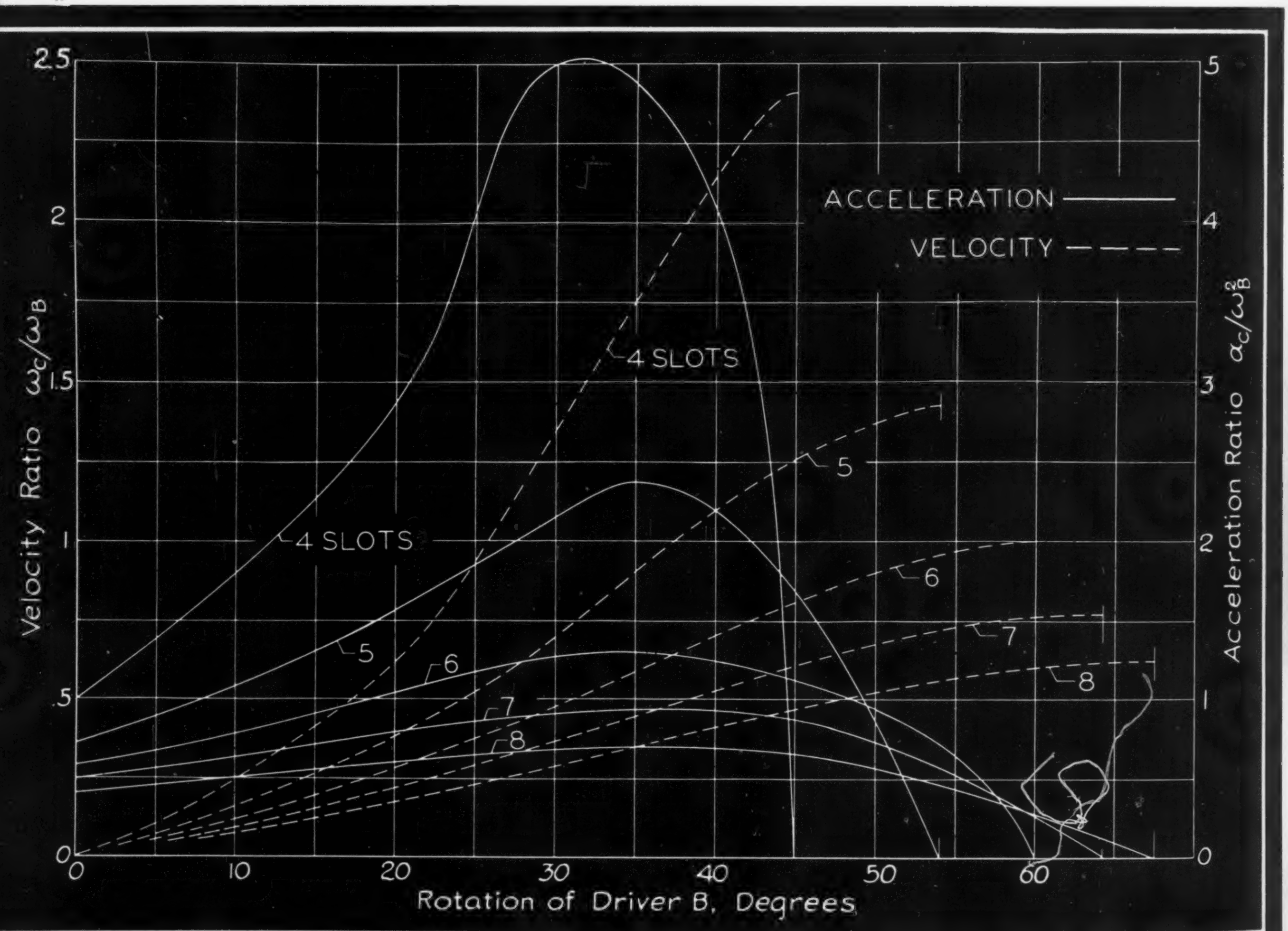
$$\cos(\beta-\theta)=\frac{1}{4}\sqrt{\left(\frac{c}{r}\right)^2+\left(\frac{r}{c}\right)^2+34}-\frac{1}{4}\left(\frac{c}{r}+\frac{r}{c}\right)$$

Substitution of this value of θ into Equation 2 gives the maximum acceleration ratio.

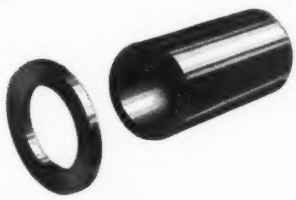
References

1. Billings, J. H.—*Applied Kinematics*, second edition, Page 303, D. Van Nostrand Co. Inc., New York.
2. Hall, H. S. and Ault, E. S.—"How Acceleration Analysis Can Be Improved", Machine Design, Feb., 1943, Page 100 and March, 1943, Page 90.

Fig. 2—Velocity and acceleration curves for driven member C of geneva mechanism for constant angular velocity of driver equal to one radian per second



Makes the TOUGH *jobs* EASY

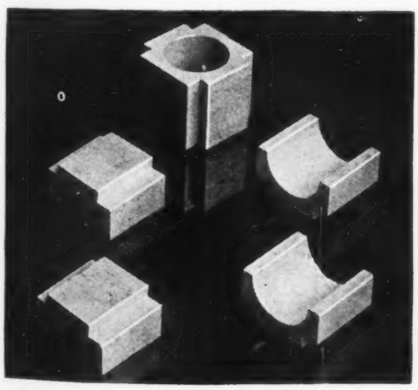


Here's a typical example of Johnson Bronze Engineering Service at work. The application in question had been served by a steel washer and a plain bearing. The difficulties present in this case were . . . in assembling—with two separate pieces . . . the constant maintenance and lubrication required . . . the tendency for the bearing to work out of place.

A Johnson Engineer reviewed the application and designed a new bearing . . . square on the outside . . . circular on the inside . . . with a flange replacing the washer. Then we produced them in LEDALOYL . . . our self-lubrication bearing material. Thus we solved all their problems at once . . . at a lower cost than the original assembly.

Excellent delivery can be had on LEDALOYL at the present . . . either from more than 2000 stock sizes . . . or made to your blue print specifications.

This is the kind of helpful service we render manufacturers every day. Perhaps we can help you with your bearing problems—NOW? There's a Johnson Sales Engineer as near as your phone.



Ledaloyl is ideal for odd or intricate shapes, as it eliminates all machining, provides self-lubrication. Write for new literature.

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SLEEVE BEARING HEADQUARTERS

The MOST COMPLETE SLEEVE BEARING SERVICE *in the* WORLD

525 S. MILL STREET NEW CASTLE, PA.

PROFESSIONAL VIEWPOINTS

". . . made a distinct contribution"

To the Editor:

Referring to the article on "Engineering Aspects of Plywood" by Ivar C. Peterson, we welcome such articles in the literature of the industry as there is far too little understanding of proper methods for utilizing plywood in aircraft design. We believe Mr. Peterson has made a distinct contribution in calling attention to the fact that the normal range of variation in wood is a factor which must be intelligently considered in any design problem. The data that he has given will be helpful to design engineers and when consolidated with others of like import should make valuable reference data for engineers. In this connection Mr. Robert W. Hess, of the Curtiss-Wright Aviation Corp., Buffalo, N. Y., recently delivered a paper before the Wood Industries Division of the American Society of Mechanical Engineers at its spring meeting at Davenport.

—Thomas D. Perry
The Resinous Products and Chemical Co.

". . . designing gears for export"

To the Editor:

I would like to draw attention to the importance of designing gears on machinery for export from a more international point of view. Even if we only go "south of the border", we come up against the module, because the meter is the official unit of measurement in Mexico, and with it the module pitch for gears.

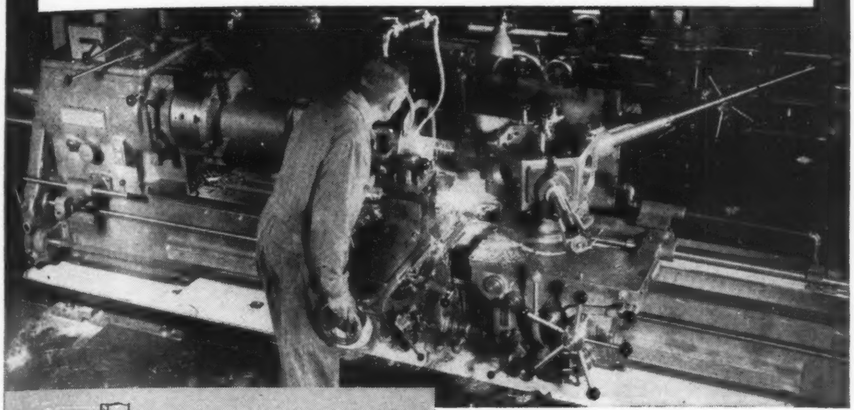
Whether intentionally or not, the inch and the diametral pitch opposed to the meter and the module, have been effective barriers to international trade. All countries outside the United States and Great Britain and her colonies and dominions use the module for gears. Russia uses it and China, as well as France and Spain and their colonies, as do Turkey and Iran. Also, our boys fighting abroad have their gear replacement problem complicated by this condition.

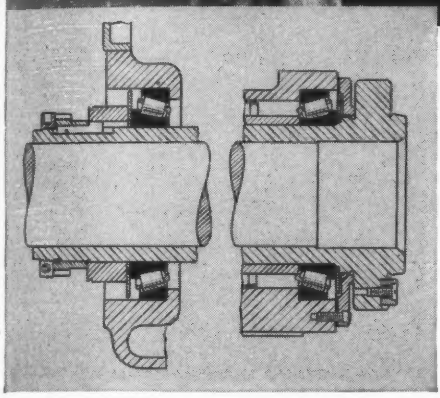
Foreign Trade Will Be Needed

With production capacity greatly enlarged now, we may need foreign trade more after the war than before, and we should do all we can to make the barrier between the two systems less formidable.

The standardization of gears on an international basis is probably still a long way off, but in the meantime we

IN THE NEWS
WITH TORRINGTON—BANTAM

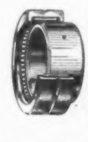




HAIR-BREADTH PRECISION WORK at high speeds is being turned out with turret lathes such as this to help win the battle of production. Built by Gisholt Machine Company, this lathe employs Ultra-Precision Tapered Roller Bearings on the spindles (see cross-section) as an aid in maintaining extreme accuracy. The eccentricity and face run-out of these bearings is .0002 maximum—an outstanding example of the ability of Bantam Bearings Division to build precision bearings for heavy-duty applications.

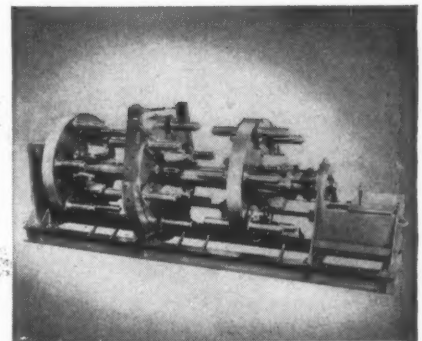
NEEDLE BEARINGS—ALL TYPES—ALL SIZES

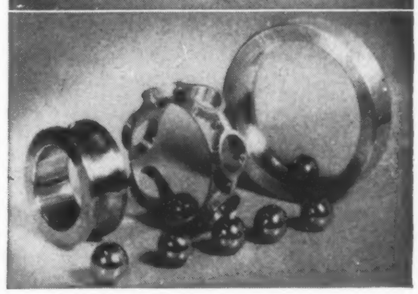
NEEDLE BEARINGS TYPE DC are complete, self-contained units consisting of a full complement of rollers and a drawn, hardened outer race. They offer the advantages of small size, low cost, high capacity—and easy installation.



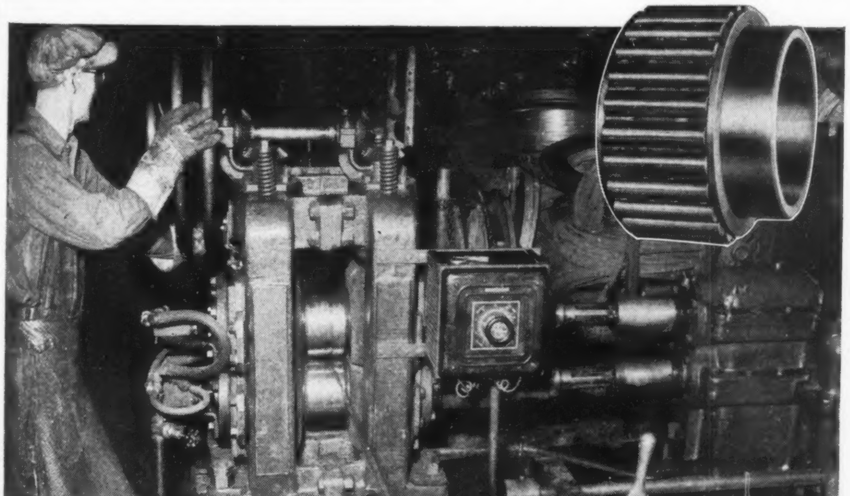
NEEDLE BEARINGS TYPE NCS consist of a full complement of rollers and a relatively heavy hardened outer race. They are furnished with or without inner races. Needle Bearings Type NCS are adaptable to heavier loads than Needle Bearings Type DC.

NEEDLE ROLLERS TYPE LN are produced in a range of types and sizes for assembly into low-cost, high-capacity anti-friction bearing units. Our engineering department will make recommendations as to correct size and type selection.





ROTATING 650 POUNDS OF WIRE AT 120 R.P.M. requires spindles of exceptionally high load capacity in wire stranding units built by The Watson Machine Company. Dependability in the main sheave bearings under this severe strain is assured by the use of Ball Radial Bearings. Designed and built by Bantam Bearings Division, these bearings have withstood years of such service without a single replacement.



HIGH CAPACITY in a small space is provided by straight radial roller bearings in this 8-inch flattening mill built by Broden Construction Company, while ball bearings, also designed by Bantam Bearings Division, take the thrust load. The bearing arrangement, shown in cross-section, permits compact design.

ARMY ★ E ★ NAVY

TORRINGTON BEARINGS

STRAIGHT ROLLER · TAPERED ROLLER · NEEDLE · BALL

THE TORRINGTON COMPANY · BANTAM BEARINGS DIVISION
SOUTH BEND 21, INDIANA

should give serious thought to equipping our machines for export to metric countries with module gears, or at least with diametral pitch gears with a center distance for which our customers abroad can, in an emergency, have gears of a module pitch made locally.

It would depend on the kind of machine and the gears used, whether to make diametral pitch gears to a millimeter center distance with oversize gears, so that they can be replaced by standard module gears as shown in the tabulation, or to make diametral pitch gears to their standard center distance and advise the customers and agents abroad of the respective module gears to use for replacement.

Diametral Pitch Gears for Replacement Modules

	Replacement Module	
Original D.P.	Special*	Standard Stock†
1	27	24
1½	18	16
2	14	12
3	9	8
4	6½	6
5	5½	5
6	4½	4
8	3½	3
10	2¾	2½
12	2¼	2
16	1¾	1½
20	1½	1¼
24	1¼	1
32	.8	¾
48	.6	½

*Special diametral pitch gears made to the standard center distance of replacement Module gears.
†Standard stock diametral pitch gears made to the standard center

If "special" diametral pitch gears are used anyhow, they could be made to a standard millimeter center distance for the respective module gears. However, as standard stock module gears are not used abroad to anywhere the same extent to which diametral pitch stock gears are used in this country, the second solution, according to the tabulation, would seem to be the most logical to use for most applications.

Incidentally, it would be a good sales point to say in literature on the machine to be exported: "The gears on this machine can be replaced by gears of module pitch."

—Fred A. Brackman
New York City

". . . interesting and instructive articles"

To the Editor:

Please send me a copy of your reprint "Modern Methods for Calculating Working Stresses", by Joseph Marin. I take this opportunity to compliment you on the interesting and instructive articles in Machine Design, and assure you I look forward each month with keen anticipation to reading the various contributions.

—Gordon J. McEachern
Toronto, Ont.

The reprint mentioned contains the entire series of articles by Joseph Marin from January to December, 1942. It is a 68-page booklet, available at a nominal price of $1.—ED.

NICKEL AIDS THE FARM IMPLEMENT INDUSTRY

to KEEP 'EM SERVING!

America won a victory on the food front in 1942 with production that reached an all-time high. But even that wasn't enough to feed a hungry, war-torn world. So this year's objectives are higher.

Farmers need mechanical equipment to produce food in winning quantities. Now, with the critical shortage of skilled farm labor, modern machinery becomes vitally important.

Agricultural equipment engineers have steadily improved farm machinery, lowered costs, and educated users. Result: no nation's farms were ever so fully mechanized as ours. And repair and service facilities have kept pace so farm equipment keeps working without excessive breakdowns or delays.

The foresight of these engineers is shown by their repeated specification of materials strengthened and toughened with Nickel. Their experience proves that a little Nickel goes a long way in the manufacture of parts that stay on the job . . . stressed parts that stand shock overloads.

From tractor transmissions to plow discs, from cylinder liners to bull gears, Nickel teams up with other metals to assure the dependable performance of farm implement parts.

For years, the technical staff of International Nickel has been privileged to cooperate with the engineers and production men who are responsible for such great achievement in the improvement of farm equipment . . . equipment now working at top speed to feed the United Nations. INCO's staff of engineers and metallurgists offers counsel and data to all who seek assistance in the selection, fabrication and heat treatment of ferrous and non-ferrous metals.

New Catalog Index

New Catalog C makes it easy for you to get Nickel literature. It gives you capsule synopses of booklets and bulletins on a wide variety of subjects — from industrial applications to metallurgical data and working instructions. Why not send for your copy of Catalog C today?

★ *Nickel* ★

THE INTERNATIONAL NICKEL COMPANY, INC., *67 Wall St., New York 5, N. Y.*

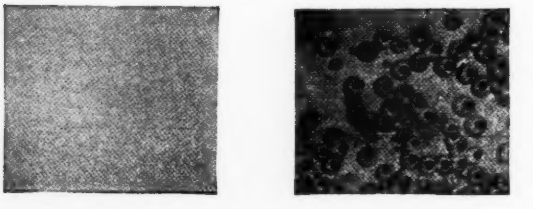
New PARTS AND MATERIALS

Outdoor Photoelectric Relay

DESIGNATED as Type CR7505-K108, a new general-purpose photoelectric relay for outdoor use where rapid and accurate counting, controlling, sorting or limiting are required, has been announced by the Electronic Control section of General Electric Co. Contacts control 2 amperes at 115 volts, 25 to 60 cycles, alternating current, or .5 amperes at 115 volts, direct current. In addition to a Type GL-930 phototube, the new relay contains a Type GE-117P7GT pliotron tube, the filament of which operates on full line voltage eliminating need for a filament transformer. Incorporated also is a diode rectifier which functions when alternating-current power supply is used. The relay's weatherproof case is equipped with a sun shield and a large, directional lens system to minimize the effect of slanting sun rays. In addition, the lens system increases the relay's sensitivity. Adjustable under actual operating conditions, without removing cover, the relay can be mounted in any position. Chassis can be removed easily from the case.

Improved Coil Coating

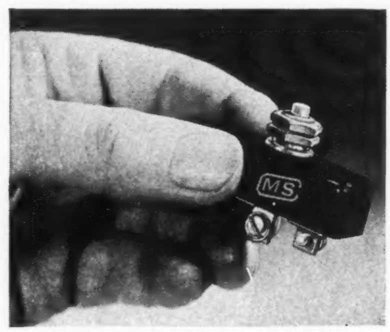
GUARDING against attack by mildew, fungi and other organisms without increasing insulation values or causing corrosive action, Insl-x Co. Inc., 857 Meeker avenue, Brooklyn, has introduced its new coil coating with improved Insl-x toxicant added as a solution to numerous tropical insulation problems. According to the



company, six-month exposure tests revealed the surface protected with the improved toxicant completely free from organic attack while a similar surface protected with ordinary toxicant was attacked. An added feature of Insl-x No. 67 (with toxicant added) is its unusual tensile strength which seals and shapes the coil and eliminates necessity of taping. The coating air dries at room temperature in 15 minutes. Dielectric strength is 1250 volts per mil.

Tiny Precision Switch

IN FULL scale production now is the new tiny Peanut switch of Micro Switch Corp., Freeport, Ill. No bigger than the end of a man's index finger, the small precision switch is more rugged, lighter and more resistant to vibration than the larger models, according to the company. Available both with and without an enclos-

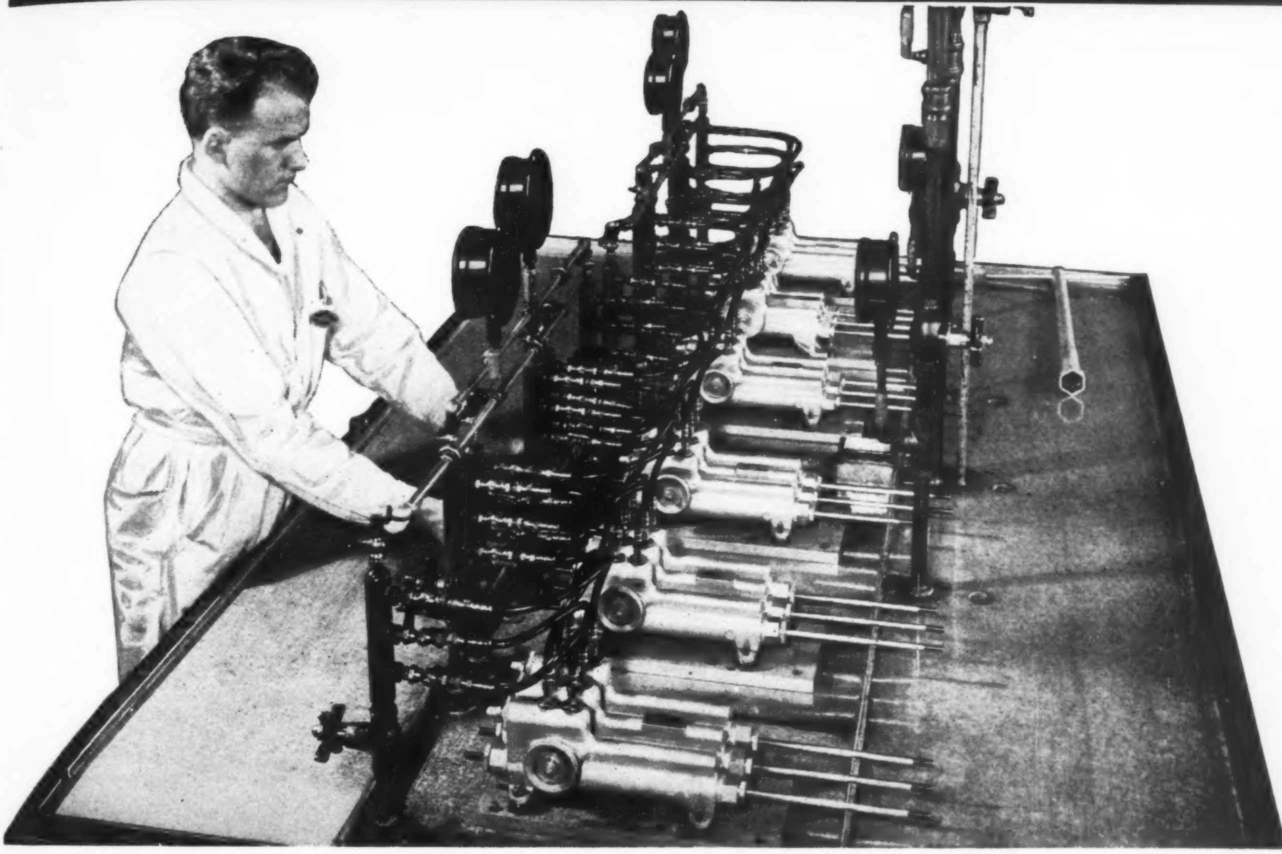


ing case, the switch possesses a high electrical rating. When used without the case, actuating movement can be applied anywhere over a large portion of the upper spring. With the case a mounting is provided. Construction provides inherent overtravel beyond the point of operation, enough for most uses. Included also are other features. Contact gaps up to .085-inch which can be varied (wide gap is especially valuable on direct-current loads), and pure silver contacts formed with a knurled surface to provide high unit contact pressure for low-voltage applications. The switch is especially suitable for use where extremely low operating force and super-fast action is not so important.

Sheaves for Multiple V-belt Drives

OUTSTANDING features of the new QD (quick-detachable) sheaves being offered by Pyott Foundry & Machine Co., 328 North Sangamon street, Chicago, are the ease, speed and safety with which the sheaves are attached to or removed from the shaft. Simple steps in attaching sheave to shaft involve slipping the tapered, split hub onto shaft in line with keyway. Headless capscrew is then tightened down with an inserted key, clamping hub firmly on shaft and producing virtual press fit. The QD sheave is attached on the tapered hub, and ac-

TESTING "ELMER'S" MUSCLES



"Elmer" is the automatic pilot into whose robot care is entrusted the flying of all types of planes. The hydraulic mechanism for the automatic pilot is the Servo unit, a number of which are shown in the above photograph on the test line at Sperry Gyroscope Company.

This Servo unit may be termed the brawn or muscle of the mechanical pilot. The pistons in the three cylinders automatically control the rudder, elevators and ailerons. A special oil pumped under 150 pounds pressure is the hydraulic medium.

VIM Leather Packings have long been used to hold oil pressure. They are not affected by the oil, because of the special impregnation which makes them oil-resistant to a much greater extent than is any synthetic type of packing.

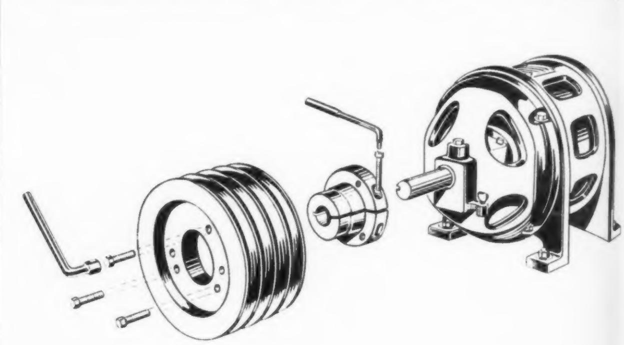
Each Servo unit has three "V" packings at each end of the cylinder and two VIM cups on the piston rod; VIM leather washers are used on the valves at the top of the cylinders.

This is another way VIM Leather Packings are serving our armed forces. They are also found in shock struts, hydraulic jacks, gun recoil mechanisms, presses, and throughout the war manufacturing industries. If Houghton can serve you on designing and engineering packings, please feel free to present your problem to our technical staff.

E. F. HOUGHTON & CO.
PHILADELPHIA
CHICAGO DETROIT SAN FRANCISCO

HOUGHTON'S
Engineered **VIM** *Leather Packings*

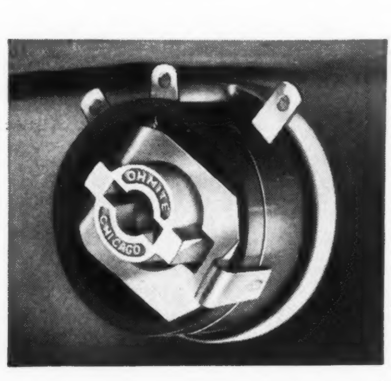
tual tightening is accomplished by inserting and tightening pull-up bolts which protrude through the sheave. Removal of the sheave from the hub is equally simple, and regardless of the number of times the process is repeated,



the mounting and demounting of the sheave is easy with no danger of a damaged shaft or battered motor bearings. A pre-aligned drive is assured as on completion of tests the hub can be left on the shaft. The new sheaves are available in standard sizes.

Rheostat-Potentiometer Available

As an addition to its instrument line, Ohmite Mfg. Co., 4835 West Flournoy street, Chicago, has designed a circular slide-wire rheostat. This low resistance low wattage rheostat-potentiometer has a length of resistance wire stretched tightly around the outside of a cylindrical core bonded to a ceramic base, and firmly anchored to two terminals. Contact to the wire is made



by a phosphor-bronze spring arm which is connected to a third terminal. Maximum resistance supplied is approximately 1 ohm while the minimum is .1 ohm. Resistance vibration is stepless since the contact arm travels along the wire from end to end. The units are made to order to suit particular requirements. Shafts for knob control or for screwdriver control are available.

Cores for Communications

For application in the communications field, Westinghouse Electric & Mfg. Co., East Pittsburgh, has made available a new line of type "C" Hipersil cores. These can be obtained in a wide range of sizes. They simplify assembly since only two or four pieces are in-

FOR WAR TODAY—FOR YOUR PRODUCTS TOMORROW

THE New Jersey zinc ALLOY POT

A publication issued for many years by The New Jersey Zinc Company to report on trends and accomplishments in the field of die castings. Title Reg. U. S. Pat. Off.

MACHINE DESIGN EDITION **No. 11**

DIE CAST INDICATING LAMPS IN WAR PLANES

Two sizes of lamps and a set of the die cast parts

Did you ever get a look inside an Army plane? The thing that impresses you most is the maze of dials, switches, indicators, etc., required to "keep 'em flying." Only the finest materials and workmanship of the electrical industry go into this equipment, and it is significant that zinc alloy die castings serve here in many ways.

Take, for example, the indicating lamps which flash operating guidance to the plane crew. Many of these lamps —which are used on Radar panels, too — are assembled with zinc alloy die castings. The lamp castings shown above, which include the housing, bezel and assembly nut, are all cast in the same die. The external threads on the housing are integrally cast and require only a simple chasing operation. And a further feature obtained in the casting operation is a tiny lug (see arrow) which prevents the indicating lamp from turning after it is embedded in the panel.

ZINC ALLOY DIE CAST CLUTCH SEGMENTS

The operating principle of the centrifugal clutch illustrated below (left) is embodied in the adjustable feature of the six zinc alloy die cast segments (right). These segments have adjustable spring-screws which keep them against the clutch hub to overcome the centrifugal force up to the adjusted speed—at which point the segments fly out and engage the clutch by a braking action on the drum. There is a facing riveted to each segment for this purpose.

The clutch drum and integral pulley rotates on two ball bearings, and a V-belt drive is used to transmit the power from an engine crankshaft to any operation desired.

By die casting the segments in zinc alloy, the necessary weight is obtained (approx. 14 oz.) and the hole for the adjusting pin is cored in the casting operation. The only machining required on the segments is the drilling of the four rivet holes for applying the brake facing.

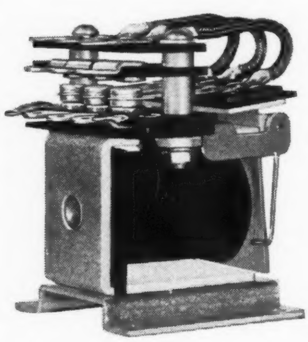
The die cast segments are adjusted to the desired clutch speed

THE NEW JERSEY ZINC COMPANY 160 FRONT ST., NEW YORK CITY

HORSE HEAD SPECIAL (99.99 + % Uniform Quality) ZINC

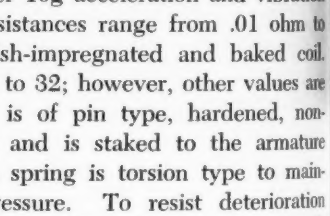
volved. Construction of the cores consists of winding the Hipersil strip continuously on a mandrel of desired dimensions. It is annealed at high temperature and vacuum impregnated with compound to make it a solid unit. It is then cut into two segments, the ends being machined and worked to produce coinciding surfaces when reassembled. For frequencies up to 400 cycles, Type C steel cores of nominal gage are used, and for those higher, a core using thinner Hipersil steel has been designed. For very high frequencies and exceptional fidelity a still thinner gage is available.

General-Purpose Radio Relay



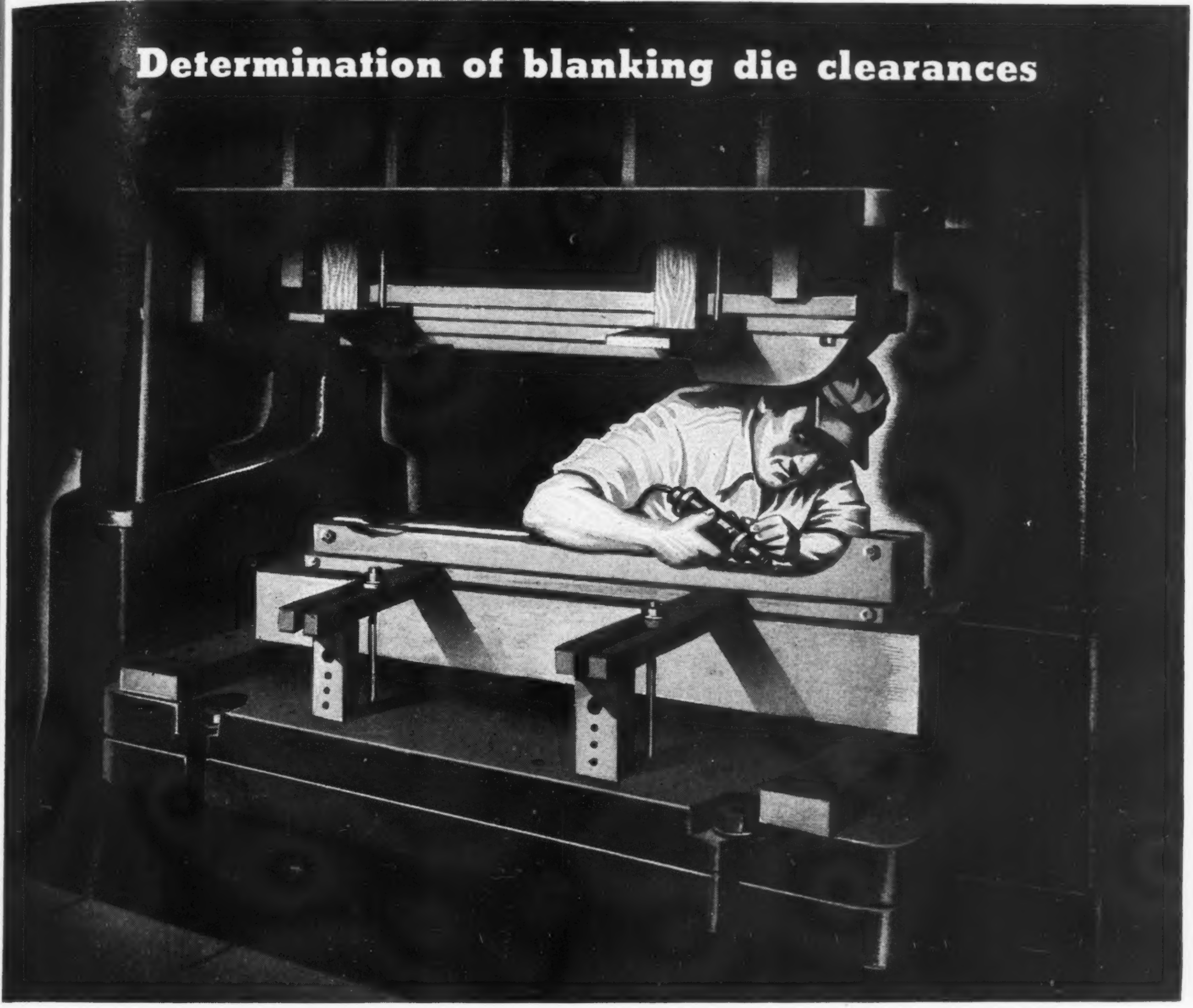
Series 345 relays designed by Guardian Electric Mfg. Co., 1601 West Walnut street, Chicago, are general-purpose radio relays available in contact combinations from single-pole, single-throw, up to three-pole, double-throw. This feature combined with the large coil winding area makes the series an efficient relay in compact space. Dimensions of the new relay are 2⅜ x 2 1/32 x 1 11/16 inches. Contacts are rated 12 amperes at 24 volts direct current, and are arranged to resist over 10g acceleration and vibration in all positions. Coil resistances range from .01 ohm to 15,000 ohms in a varnish-impregnated and baked coil. Standard voltages are 16 to 32; however, other values are available. The bearing is of pin type, hardened, nonmagnetic, stainless steel and is staked to the armature hinge. Armature return spring is torsion type to maintain an even spring pressure. To resist deterioration under conditions of high humidity, relay parts are plated.

Aircraft Cowling Fastener



Known as the "Jiffy Fastener" for aircraft cowlings, inspection plates, protective panels and similar uses, a new fastener is in production now by Simmons Machine Tool Corp., Albany, N. Y. It features a helical-type spring which gives long spring travel, reduces vibration and assures a tight, rigid fit. In tension the load is not carried by the spring but by hardened steel lugs. The sole function of the spring is to hold the two parts of the fastener in a locked position. Another improvement is the fact

Determination of blanking die clearances



Information supplied by an Industrial Publication

Determination of proper blanking die clearance depends on several factors; thickness and physical properties of stock, relation of punch diameter to stock thickness, specified part tolerances and press power and size.

Recommendations of material suppliers regarding clearances for every type of blanking operation can usually be followed. Lacking recommendations, or when clearance must be determined by experiment certain simple rules give reasonably accurate results.

The amount of clearance varies from 5 to 12% in direct proportion to the stock thickness. Closer tolerances call for smaller clearances. The following table gives general average total clearances.

	Close Tolerance	General Run
Brass and Soft Steel	5%	8%
Medium Rolled Steel	6%	10%
Hard Rolled Steel	5-7%	12%

When the blanking or piercing hole must be held to a close tolerance, clearance is added to the punch dimensions. When the blanked part must be held to close tolerance, clearance is subtracted from the dimensions.

CLIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

ARMY E NAVY

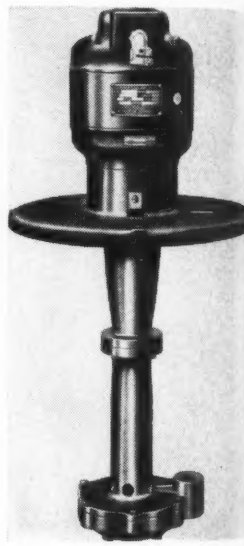
MOLYBDIC OXIDE, BRIQUETTED OR CANNED • FERROMOLYBDENUM • "CALCIUM MOLYBDATE"

MOLY

Climax Mo-lyb-den-um Company
500 Fifth Avenue • New York City

that the stud is self-ejecting when unlocked and can be readily recognized as being in that position during repairs or re-assembling work. Stud part is also locked semipermanently in outer sheet and thus cannot be lost or mislaid during removal of the cowling. The fastener is self-aligning after first installation. No vibration occurs when the fastener is in unlocked position. The tapered design of the fastener permits ample clearance for assembly of cowlings. It can also be used on curved sheets of practically any radius. Deflection is held to a minimum. All wearing parts are heat-treated and the entire assembly is cadmium-plated for corrosion resistance.

Deep Immersion Coolant Pump



IDENTIFIED as Model "TL", a new vertical "gusher" coolant pump has been designed by The Ruthman Machinery Co., 1810 Reading road, Cincinnati, for deeper immersion. It is capable of handling soluble coolants containing grit and abrasives without injury to the mechanism. An integral mounting flange is provided which is usually bolted to the reservoir cover or bedplate, forming a simplified installation in connection with a comparatively deep reservoir which is frequently arranged below floor level. No packing glands or friction seals are used, nor are there any bearings or metal-to-metal contacts below the mounting flange. The one-piece shaft rotates on two large ball bearings and extends from the motor through the tubular housing to the pump impeller. One bearing is mounted in upper motor end bell and the other within the mounting flange housing, providing sufficient distance between bearings to maintain rigidity in the heavy shaft extension. Weight and thrust of vertical shaft is taken by large lower bearing which is locked into position. This feature eliminates any change in the position of the impeller both before and after the motor reaches its normal operating temperature. Equipped with ½-horsepower totally enclosed motor for continuous duty, and ¾-horsepower motor for extra-heavy duty, the pump is available in two lengths from the mounting flange to the bottom of the pump, one being 23⅝ inches and the other 19⅝ inches.

Synthetic Rubber Hose

KNOWN as Hycar OR-25, a new synthetic rubber has been developed by Hycar Chemical Co., 335 Main street, Akron, O., for flexible hose and other aircraft parts for extremely low temperature conditions. The material

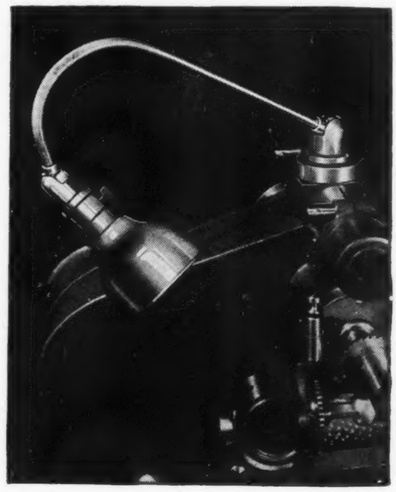
(*Continued on Page* 174)

(*Continued from Page* 170)

possesses maximum oil resistance and good tensile strength, resilience and heat resistance. Some of the uses of this flexible hose are for fuel and oil hose, coolant hose, hydraulic hose and accumulator bags which must "breathe" as they follow the variance of pressure to actuate hydraulically operated mechanisms; diaphragms for carburetors; fuel gages and valves.

Adjustable Lighting Unit

UNAFFECTED by machine vibration the new "Swivelier" machine-lighting unit of Reliance Devices Co. Inc., 510 Sixth avenue, New York, stays put at any angle, with no set screws or wing nuts to adjust. It will not work loose or drop down because of patented spring construction inside the swivel base and swivel socket. Swivel base and socket each have a wide range of movement; the socket adjusts to 90-degree vertical range and rotates

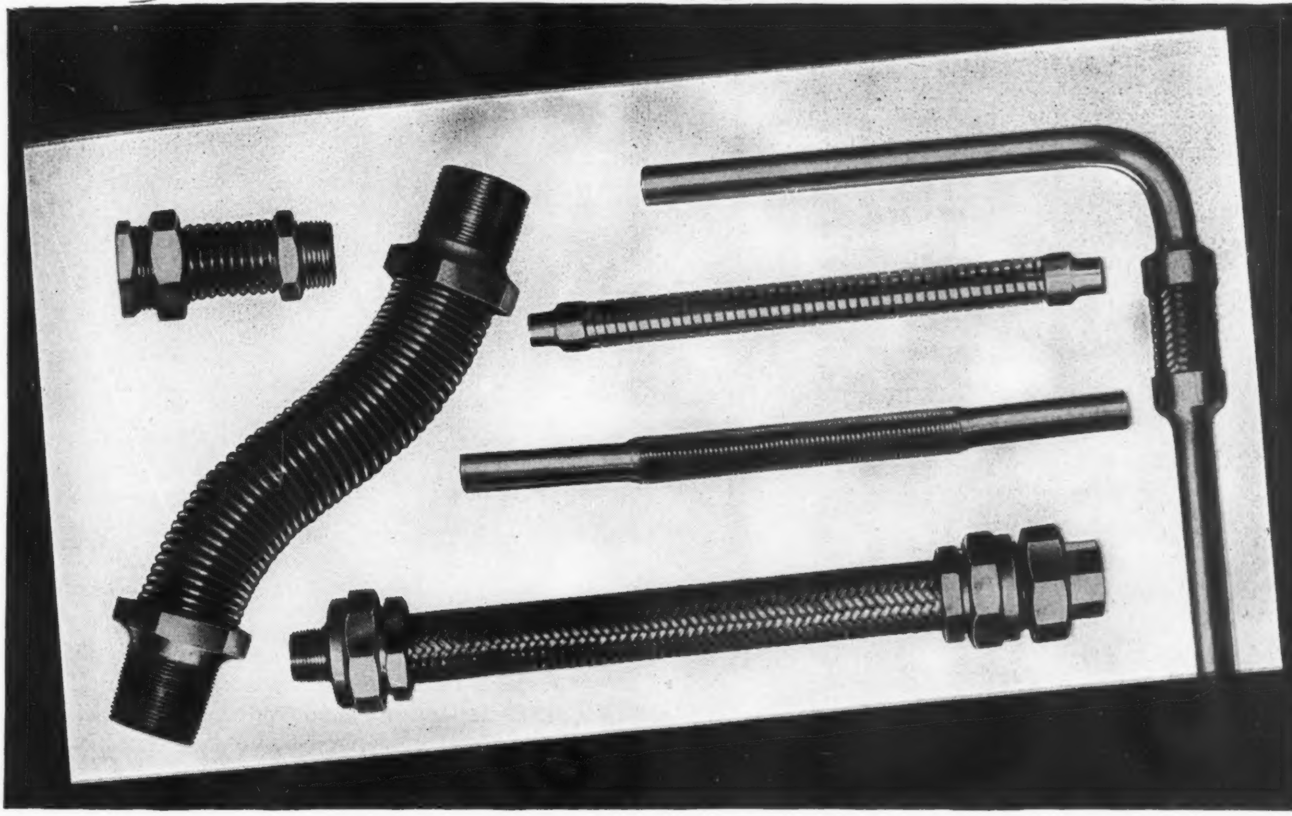


350 degrees horizontally, and the arm in base also has a 90-degree vertical adjustment and 350-degree horizontal range. The spring tension construction holds the complete unit in place at any angle, putting light on the spot where needed. Wire is completely enclosed in unit, which is oil dust resistant. A choice of arm lengths, base and socket combinations, with switch in base and keyless socket, or switchless base and key socket is available. There are also three types of shades.

Synthetic Rubber Developed

CHEMURGIC rubber developed from vegetable oils —known as Witcogum—is claimed by Wishnick-Tumpeer Inc., 40 East Forty-first street, New York, to be comparable to rubber in many of its properties, yet not requiring critical materials or equipment for its production. Material is now being used for hose, tubing, wire insulation, gaskets, shims, brake linings, and similar parts. It contains accelerator of guanadine type and sufficient sulphur to give cure in 30 minutes at 40 pounds steam pressure. All necessary vulcanizing ingredients are already present in material, though it may be loaded or

"IT'S THE MOST RELIABLE METAL TUBING WE KNOW OF"



They all agree that it is . . .

And when production-conscious executives must have *absolute, 100% tightness* plus long life in flexible metal tubing, they usually specify American *Seamless* Flexible Metal Tubing. For past experience has shown them that this deluxe product will flex millions of times without failure.

American *Seamless* is all-metal, made from *seamless tubes*. It has neither joints, welds, laps, seams nor packing at which leaks might develop. It combines the flexibility of rubber hose, the dependability of metal, the strength of rigid pipe.

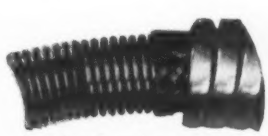
Another interesting feature—you can get American *Seamless* Tubing in practically any workable metal of your choice.

Throughout industry, American *Seamless* is almost a password. Countless war plants are using it to convey air, fuel, water, chemicals, cutting compounds, refrigerants, propane and butane, manufactured freon, ammonia, oxygen, hydrogen, nitrogen, acetylene and many other fluids. They use it too for controlling vibration and for connecting moving or misaligned parts.

Whatever your requirements you'll most likely find that we have a flexible metal tubing or hose that will help you do the job just a little bit better.

Why not investigate?

43200



American Seamless—corrugated from seamless rigid tubing . . . pressure tight as the metal tube from which it is made.

ANACONDA

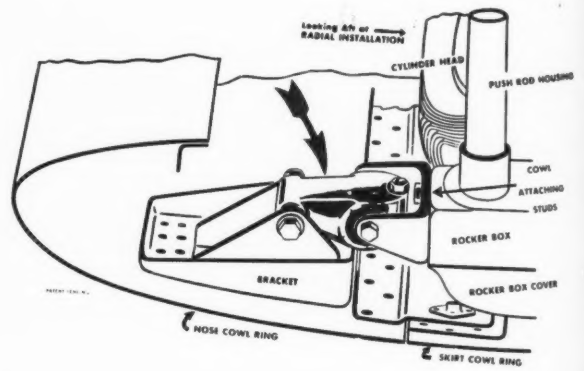
American Metal Hose

AMERICAN METAL HOSE BRANCH OF THE AMERICAN BRASS COMPANY • General Offices: Waterbury, Conn.
Subsidiary of Anaconda Copper Mining Company • *In Canada:* ANACONDA AMERICAN BRASS LTD., *New Toronto, Ontario*

softened as requirements demand. It may be used independently or as extender blended with natural rubber, reclaimed or synthetic rubber. Tensile strength of 450 pounds per square inch, elongation as high as 150 per cent, shore hardness of 6065 and tear of 45-50 pounds per inch can be obtained through proper compounding. Water, alcohol and lubricating oils have no apparent effect.

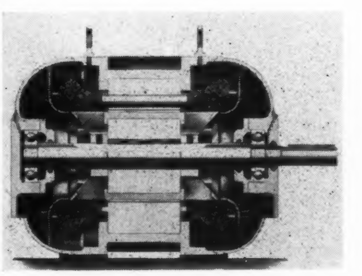
Flexible Aircraft Connector Link

EMBODYING two new and improved features, the engine cowl ring connector link for aircraft announced by Kinney Engineering Co., 2019 Bay street, Los Angeles, incorporates a rubber bushing which absorbs engine and flight vibration together with a simple rocker arm action or compensating principle that takes up normal engine



heat expansion. These features make possible a flexible, vibration-absorbing link of unusual strength that eliminates split cowling failures caused by excessive vibration transmitted from the engine to the cowling by the rigid type of connector link. The link is molded of aluminum alloy, heat-treated under continuous metallurgical control, and X-ray tested. It is completely machined and assembled and has an anodized finish.

Drip-proof Induction Motor

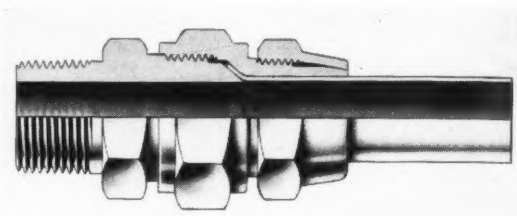


OF sturdy construction, the RS drip-proof induction motor of Lima Electric Motor Co., 2001 Findlay road, Lima, O., is built in sizes from ½ to 75 horsepower, 2 or 3-phase, alternating current. Designed to afford maximum protection against chips and filings, dripping or splashing liquids, it is adaptable to machine tools and equipment for canneries, paper mills, dairies, packing houses and other locations where a totally-enclosed motor is desirable. The motors are equipped with heavy-duty

ball bearings, heat-treated and ground shafts, dynamically balanced rotors, and heavy rugged frames. Phase insulation is heavy varnished duck.

Tube Fittings for Hydraulic Systems

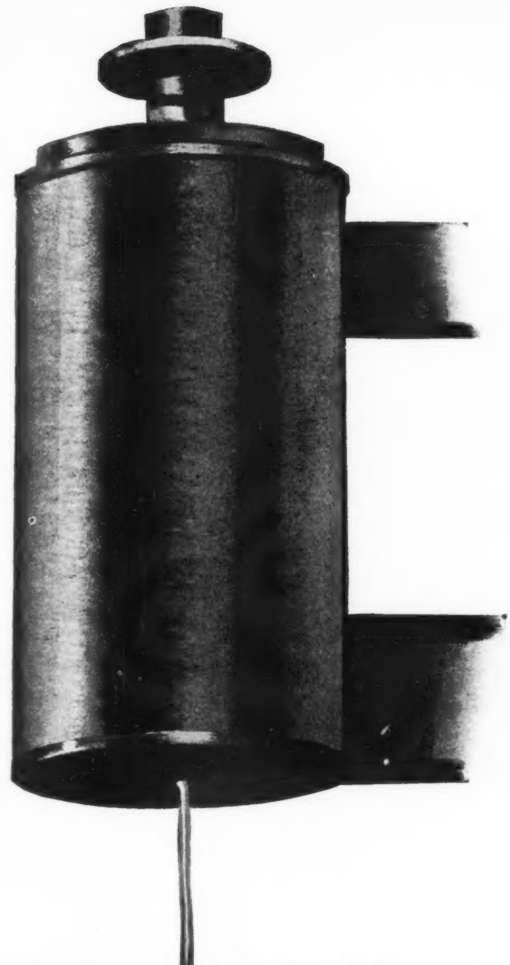
HYDRAULIC tube fittings for industrial use are being distributed by The Cotner-Wilkinson Co., Logansport, Ind., under the tradename of "Collet Grip", which describes the principle used in the fittings to provide dependable connections without the necessity of threading, welding or soldering. The double-nut series shown in the illustration is available in many models for tube sizes ranging from ¼-inch to 2 inches. Compression nut compresses the collet nut to tubing, giving a firm grip that



will not loosen under vibration or shock. This nut also compensates for any variation in the outside diameter of the tubing. The collet nut, with its long bearing surfaces, grips tube firmly and securely without damage. This "Collet Grip" design places load on the tube, directing pulling stresses and vibration strains away from the flare. Another series—the single-nut—retains all the double-nut features but is much simplified for close work. It is made for tube sizes ranging from ¼-inch to 1 inch inclusive. Short tube installations are practical with this nut, which is also adaptable where space is limited.

Quick-Releasing Hinge

DEVELOPED originally for ordnance use, a new type of quick-release hinge also has a wide potential range of commercial applications. Newly introduced by Burklyn Co., 3429 Glendale boulevard, Los Angeles, this new type of device combines a hinge and a means of instantly releasing the hinge for quick and easy removal of a door, hood, chute or other part. It is composed of a bracket housing spring-loaded attaching pins, the latter being equipped with finger pads to retract the pins. When pads are forced together, which can be done from any position even with gloved hand, the hinged part is entirely free. As mentioned above, the hinge was first built to replace former methods of releasing ammunition chutes on aircraft machine guns,



Danger! CORDITE

—and how Hele-Shaw Fluid Power Engineers tamed it

The extrusion of cordite is a ticklish job. Every conceivable precaution must be exercised to protect life and property.

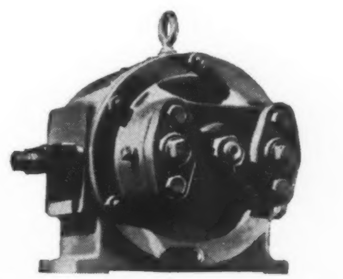
Our engineers were asked to cooperate. Because of the safety factors there were many complications.

It was desirable, for example, to bring the ram down to the plug quickly to avoid wasting time. However, once the ram neared the cordite plug it had to stop, proceed automatically, and contact the plug without shock. There was still another condition. Once contact had been made, speed of extrusion and pressure had to be automatically controlled, depending on the consistency of the plug.

After conferences with the manufacturer, we designed and furnished the entire pumping unit and oil circuit, including Hele-Shaw Fluid Power pumps and special controls.

Your post-war thinking may lack the drama of a cordite extruder, but offer equally baffling problems in improving a product or process, or in simplifying control or operation of a machine. If you have an idea Hele-Shaw Fluid Power or Hele-Shaw engineers can help you in your post-war planning, you are invited to write for further assistance.

THE
Hele-Shaw
Fluid Power Pump



ÆCO 75th ANNIVERSARY

OTHER A-E-CO PRODUCTS: TAYLOR STOKERS, MARINE DECK AUXILIARIES, LO-HED HOISTS

AMERICAN ENGINEERING COMPANY

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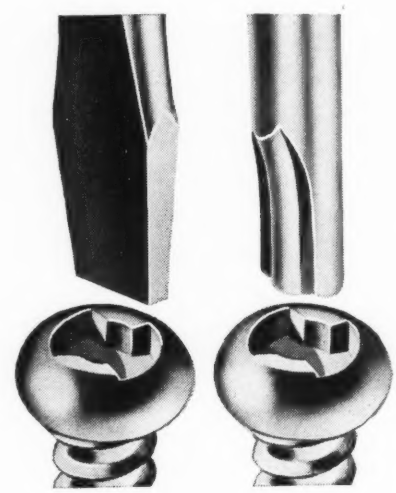
and now finds applications on machine tools, generator housings and automotive equipment, on marine hatch covers, cowlings, folding seats and tables in aircraft, cabinet lids and wherever hinged hoods or surfaces must be occasionally removed for cleaning or service. Available in several bracket designs and lengths from 2 to 6 inches, the hinge is now in large scale production in noncritical materials.

Clutch-Head Screws Developed

ADAPTABLE to a wide range of head designs is the new clutch head used by United Screw & Bolt Corp., 2515 West Cullerton street, Chicago, in its recessed head screws. Under various assembly conditions, according to the company, the new clutch head screws have proved their merits. Either an assembly bit or a standard type screwdriver may be used. Driver does not



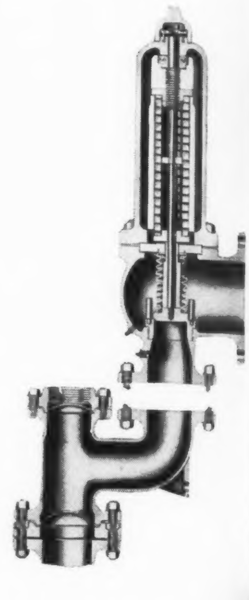
slip out of the clutch recess so that finished surfaces do not become scratched. For initial assembly and conditions where screws are hard to get at or where a one-hand drive is required, an assembler's bit is preferable. Screw and bit lock together by slight twist of screw, or bit, reducing possibility of dropping screws and saving time in assembly. Lock releases instantly when driving of screw starts.

All-Leather V-Belt

AN ALL-LEATHER V-belt has been announced by The Charles A. Schieren Co., 37 Ferry street, New York. The design of the new belt is said to combine the pulling power of a flat leather surface and the grip of the V-drive. Full grain leather sides are firmly cemented to a leather core, which combines the strength of leather and the power transmission qualities of full grain leather side walls in contact with pulley sheaves. The new design also permits belt to run slack. Tests over the past two years have indicated that more revolutions per minute were provided, even in cases where belts were oil soaked, and the stretch was not worthy of notice. The V-belts

are supplied in rolls of the three popular standard sizes A, B and C. Simple portable devices are provided to make the belts endless either when on or off the pulleys.

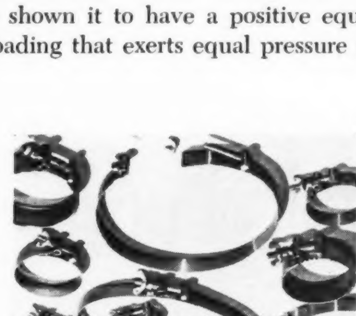
Pressure Protection Unit



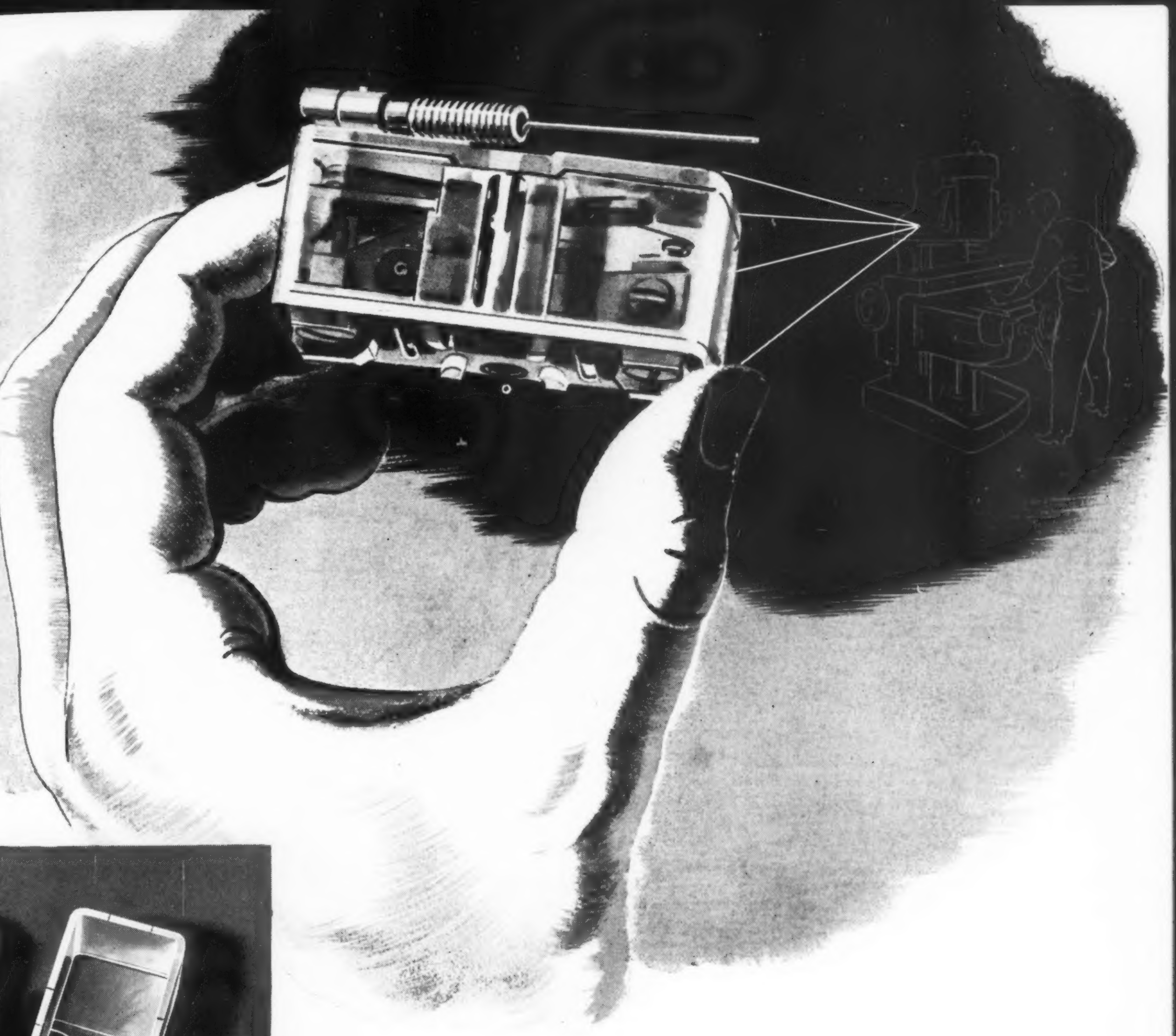
FOR use in synthetic rubber plant equipment or wherever corrosive or viscous chemicals warrant use of a rupture diaphragm beneath a relief valve, Black-Sivalls & Bryson Inc., Kansas City, Mo., is offering complete pressure protection in its Master Assembly relief valve. It combines the primary and emergency relief devices in one unit. A safety head isolates relief valve from corrosive or gummy contents of vessel. When diaphragm of safety head is burst by overpressure, "conservation" relief valve blows until pressure drops to normal, then closes to conserve contents of vessel. If relief valve proves inadequate and pressure continues to rise, diaphragm of emergency safety head bursts before pressure reaches danger point. It is suggested that setting of safety head and relief valve should not exceed maximum allowable working pressure of vessel.

Air and Oil Duct Clamp

TO MEET production demands of aircraft manufacturers, Marman Products Co., 940 West Redondo boulevard, Inglewood, Calif., has developed a revolutionary type of air and oil duct clamp. It is claimed that tests have shown it to have a positive equalized circumferential loading that exerts equal pressure at every point,



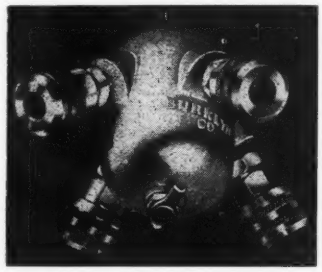
and it can be used over and over as its clamping efficiency is maintained indefinitely. Built especially for high-pressure hot or cold installations for aircraft, automotive, tank or marine engines and assemblies, the diameter sizes of the clamp, ¾-inch to 38 inches, conform to any convex



surface whether square, oblong, circular or triangular. The clamp is strong and light in weight, can be easily installed and is adaptable to many uses. They are available from stock, in sizes mentioned, in various metals such as dural and aluminum, and cold-rolled, cadmium-plated, and stainless steel. Other sizes and types are built to specifications.

Four-way Air Manifold

AVAILABLE for immediate delivery is a new four-way air manifold of Burklyn Co., 3429 Glendale boulevard, Los Angeles. The manifold is a suspension type which accommodates up to four air hoses. It has a water trap and drain to reduce moisture from compressed air reaching the air tools or jet outlets, and may be used with quick-detachable hose fittings or with straight



screw-in connections. Intake is ½-inch pipe thread and the four outlets each are ¼-inch pipe thread. The manifold has found application in sandblasting machines, spray-painting equipment and other industrial operations where extra air hose connections are needed from one main pipe or hose outlet.

Engineering Dept. Equipment

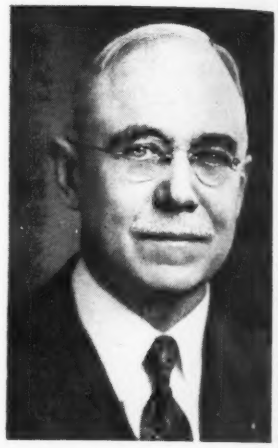
Redesigned Fluorescent Fixtures

REDESIGNED to WPB requirements, the Generalite fluorescent fixture line is now being offered by The Fostoria Pressed Steel Corp., Fostoria, O. The nonmetallic reflectors are removable without turning off current or removing bulbs. V-shaped wiring channel separates the two 40-watt lamps and acts as reflector. Three four-lamp units are available for continuous line or individual lighting applications. Model MF-240 uses two 40-watt lamps and model MF-2100 uses two 100-watt lamps, both being equipped with conventional ballasts.

Moisture-Resistant Tracing Cloth

RECENTLY developed by The Frederick Post Co., Box 803, Chicago, is an improved white pencil tracing cloth, Whitex. The outstanding feature of the tracing cloth is its moisture resistance on both sides. A hard pencil can be used to leave a sharp, jet-black line on the fine-tooth surface. This feature plus the glass-like transparency of the tracing cloth assures sharp prints.

MEN OF MACHINES



VICE president of the American Society for Testing Materials for the past two years, Dean Harvey has been elected president of the society. He is a materials engineer at Westinghouse Electric & Mfg. Co., East Pittsburgh, and has been connected with the company for 39 years. Since 1941 he has averaged three to four days weekly in Washington as a dollar-a-year man, serving as section chief of the Conservation Division of the War Production Board. In this capacity he has directed the revision of government specifications for electrical and mechanical equipment to provide for the conservation of critical materials. Born in Chicago, Mr. Harvey is an electrical engineering graduate of Armour Institute of Technology. Upon graduation he joined the Underwriters Laboratories as electrical engineer and later went to Westinghouse. His activities within the A.S.T.M. have been many and varied and, in addition to his wide background in materials, will enable him to perform in an outstanding manner his duties as president of the society.

NOMINATION of Nevin E. Funk, vice president in charge of engineering, Philadelphia Electric Co., Philadelphia, as president of American Institute of Electrical Engineers has been announced recently. A native of Pennsylvania, Mr. Funk was graduated from Bloomsburg State Normal School and Lehigh university. After graduation he served an apprentice course with Westinghouse Electric & Mfg. Co., and later became con-

nected with the New York Central Railroad. Prior to entering the employ of the Philadelphia Electric Co. in 1907, he was assistant professor at the Georgia School of Technology. Mr. Funk's association with the Philadelphia company has been marked with consistent advancement from his first job to vice president, the position he has held for twelve years. For many years he has been prominent in activities of various technical and engineering societies, notably the American Institute of Electrical Engineers and the American Society of Mechanical Engineers, and is a Fellow in each organization. He has served as director of the institute and on a number of its national committees, including chairmanship of the Lamme and Edison medal committees. To his credit also are 43 technical papers which he has presented at local or national society meetings, including the Second World Power conference in Berlin.

♦

Roe S. Clark has been nominated president of the National Metal Trades association; H. H. Kerr, vice president; and George A. Seyler, second vice president and treasurer. Mr. Clark is vice president and treasurer of the Package Machinery Co.; Mr. Kerr, president of the Boston Gear Works; and Mr. Seyler, vice president of the Lunkenheimer Co., Cincinnati.

♦

J. Farkas has joined Commonwealth Aircraft Inc. as engineering manager. He is well known in the aviation industry as a production expert and an engineering test pilot, and has been connected with aircraft since 1924.

♦

Dr. Willard Henry Dow has been selected as the Chandler lecturer and medalist for 1943 for Columbia university. Dr. Dow is president of the Dow Chemical Co. and has made some outstanding "achievements . . . in producing bromine and magnesium from sea water and of synthetic plastics and rubber."

♦

R. W. Lytle, associated with Formica Insulation Co. for twenty years, has been elected vice president in charge of special engineering.

♦

James M. Shoemaker, formerly assistant engineer, has been appointed chief engineer at Chance Vought Aircraft. Paul S. Baker, chief test pilot, is now engineering manager. Mr. Shoemaker joined the company in 1934 as project engineer, was named chief project engineer in 1941, and last November was promoted to the post of assistant engineer. Mr. Baker has been with the company since 1930.

♦

Dr. Walter Savage Landis has been awarded the gold medal of the American Institute of Chemists "in recognition of his contribution to chemical engineering and development work". He is vice president of American Cyanamid Co.

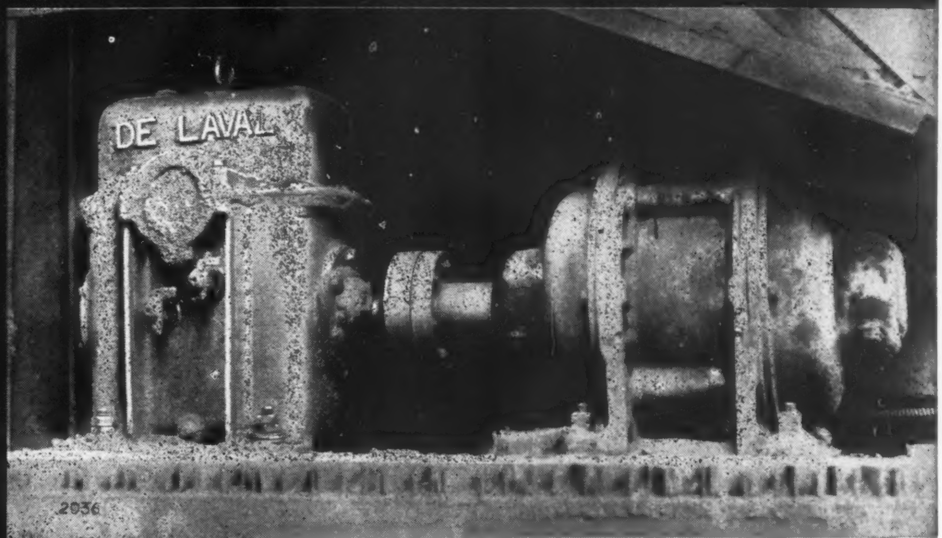
♦

Dr. Robert M. Burns of Bell Telephone Laboratories was elected president of the Electrochemical Society.

♦

J. Paul Ahlbrandt has been made chief engineer of Midwest Mfg. Co., Galesburg, Ill. For the past fifteen months he had been connected with the Office of War

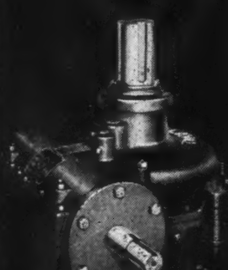
for CONTINUOUS PRODUCTION



De Laval worm gear driving coal conveyor

Bottom drive

Top drive



Gear shaft vertical

Double reduction

Breakdowns in power transmission in 24-hour plants are eliminated where plant engineers specify the

DE LAVAL
WORM GEAR

for heavy duty drives, particularly in locations involving exposure to grit, dust and moisture.

The two moving parts of the gear run in a bath of oil and it is self-lubricating, requiring no attention beyond inspection of the level in the oil well at long intervals.

The unit does not throw grease or oil.

Safety guards are unnecessary, as the working parts are enclosed.

The efficiency is high, and improves with use.

The drive is smooth and quiet. It is also positive; there is no slippage, regardless of starting torque.

Describe conditions and ask for Publication W-1132.

DE LAVAL WORM GEAR DIVISION

of the De Laval Steam Turbine Co., Trenton, N. J.

MANUFACTURERS OF TURBINES STEAM HYDRAULIC PUMPS CENTRIFUGAL PROPELLER ROTARY DISPLACEMENT, MOTOR-MOUNTED MIXED-FLOW, CLOGLESS, SELF-PRIMING CENTRIFUGAL BLOWERS and COMPRESSORS GEARS WORM HELICAL and FLEXIBLE COUPLINGS

Utilities, War Production Board, and previously with Ingersoll Steel & Disk division, Borg-Warner Corp., for eight years.

CARLTON E. STRYKER has been appointed chief of the Conservation and Standards division, Resources Control Office, Aircraft Production Board.

PROF. F. J. LINSENMEYER has resigned as director of mechanical engineering, University of Detroit, after 17 years as a member of the faculty. His new position is that of chief engineer, National Stamping Co., Detroit.

VINCENT BENDIX is the new chairman of the Langley Aviation Corp., New York, organized to develop molded plastic plywood process, design and construction methods for mass production of aircraft and parts. Mr. Bendix is a former Society of Automotive Engineers president, and is a well known automotive and aeronautical design and manufacturing executive.

CHARLES S. MCCANN, formerly assistant supervisor of engine testing, Aviation Engine plant, Buick Motor division, is now director of research of Walker Mfg. Co., Racine, Wis.

COL. JOHN H. JOUETT has been made president of the Bellanca Aircraft Corp. He formerly was evecutive vice president, Higgins Aircraft Inc.

LEONARD A. COLSTON, a designer at North American Aviation Inc., Inglewood, has been made contract engineer with Lockheed Overseas Corp., and is stationed at a foreign base.

MILES E. JOHNSON has accepted a position as engineer for Draper Motors, Detroit. His previous connection was that of division engineer with Continental Aviation and Engineering Corp.

G. ALLEN CREIGHTON has joined the Lycoming division The Aviation Corp., Williamsport, Pa., as assistant experimental engineer. He has resigned as assistant to the chief engineer, Electric Boat Co., New London Ship & Engine Works.

G. E. GUSTAFSON, chief engineer since 1933 and assistant vice president since 1940 of Zenith Radio Corp., Chicago, has been appointed vice president in charge of engineering.

C. C. HERMANN, formerly chief engineer, has been made general manager, Claude B. Schneible Co., Chicago.

R. D. SPEAS, who prior to his present appointment had been a research engineer, has become assistant to the vice president of engineering, American Airlines Inc., La Guardia Airport, Jackson Heights, L. I., N. Y.

J. J. WALLACE has become acting assistant chief engineer at the General Aircraft Corp., Astoria, L. I., N. Y. He had been connected with Bristol Aircraft division,

Universal Moulded Products Corp., Bristol, Va., as assistant chief engineer.

♦

BENHAM S. POND, who had formerly been assistant chief engineer, Acrotorque Co., Stamford, Conn., has joined the Dowty Equipment Corporation's special development division, of the same city, in the capacity of director of this division.

♦

FRANK E. MCGARY, engineering and mechanical division chief, has been appointed manager of the Murray Corp., Scranton, Pa., plant now under construction. O. F. GRAEBNER, chief engineer, replaces Mr. McGary at the main division of the Murray plant in Detroit.

♦

ROBERT INSLEY has been made vice president and executive engineer of Continental Motors Corp. He formerly was vice president in charge of engineering at Menasco Mfg. Co. Prior to this—in 1928—Mr. Insley was connected with Continental and helped organize the company's aircraft division.

♦

GEORGE H. KENDALL, consulting mechanical engineer, Norma-Hoffman Bearings Corp., has become associated with Sargent & Co., New Haven, Conn., as chief engineer.

♦

LEONARD TROY has been appointed vice president in charge of engineering at Strickland Aircraft Corp., Topeka, Kan. He had been associated with Snead & Co. in the Aeronautics division.

♦

E. B. GEORGE has become vice president and director of engineering, Leland Electric Co.

♦

DELMAR WRIGHT has been made director of a new radio and electronic research department of Bendix Aviation Ltd.

♦

A. L. POMEROY, formerly automotive engineer, has become test engineer with Ranger Aircraft Engines, division of Fairchild Engine & Airplane Corp., Farmingdale, L. I., N. Y.

♦

HAROLD E. KOESTER has been promoted chief engineer from assistant chief engineer of Righter Mfg. Co., Burbank, Calif.

♦

DONALD M. BERGES who previously had been chief engineer at Eclipse Aviation division, Bendix Aviation Corp., has joined the Pump Engineering Service Corp., Cleveland, to serve in a special engineering capacity.

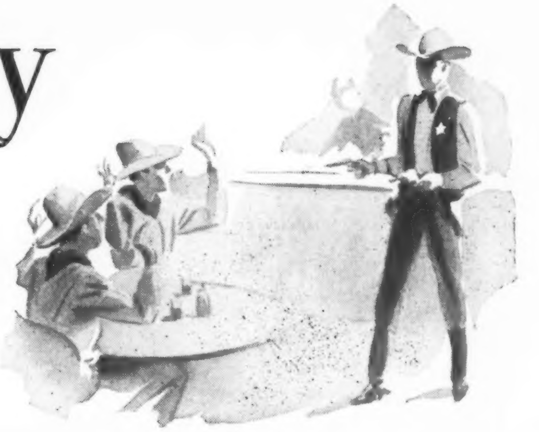
♦

R. H. LUSCOMBE, of Penn Electric Switch Co., Goshen, Ind., has been appointed president of the Refrigeration Equipment Manufacturers association. A. B. SCHELLENBERG of Alco Valve Co., St. Louis, becomes vice president of the organization.

♦

R. W. LYTLE, associated with Formica Insulation Co. for twenty years, has been elected vice president in charge of special engineering, including automotive and aircraft engineering. D. J. O'CONNOR becomes assistant chief engineer.

Colt Quality



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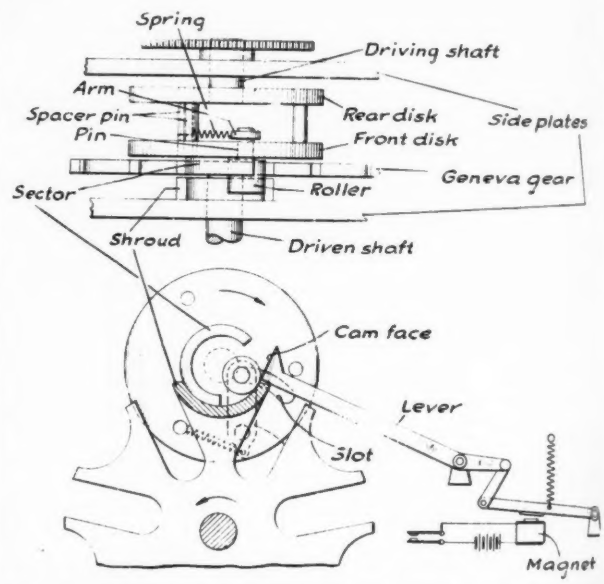
COLT'S PATENT FIRE ARMS MFG. CO., ELECTRICAL DIVISION, HARTFORD, CONN.

NOTEWORTHY PATENTS

Clutch Operates with Geneva Mechanism

IN MECHANISMS employing geneva gearing, it may be necessary to disengage the driving part from the driven in order to perform certain operations. In any such mechanical disengagement the driven part must be held and locked in a predetermined position so that the driving part may be reengaged at a later time without interrupting the orderly sequence of motion. A clutch mechanism for this purpose, which requires no critical adjustments and is of simple construction, is covered by patent 2,307,-112, recently assigned to International Business Machines Corp.

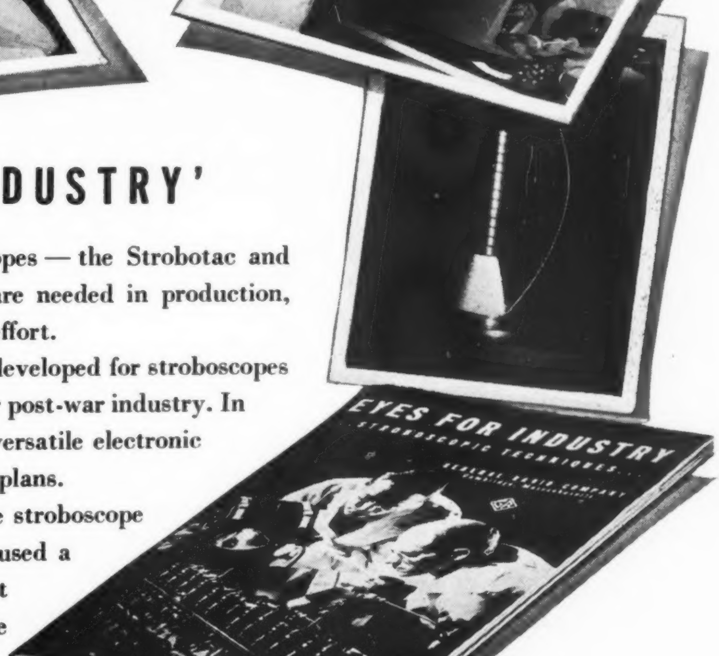
Referring to the illustration, continuous motion is sup-



Clutch mechanism is shown disengaged so that geneva gear is held locked. When magnet is energized lever is raised, permitting roller to enter slot in geneva gear

plied to the driving shaft by a gear, the shaft being supported by a bearing in the side plate. A disk fastened to the end of the driving shaft supports a second disk secured to it and spaced from it by three spacer pins. An arm pivoted on the rear face of the front disk carries a pin which projects through a slot in the disk. The slot is in the form of an arc and limits the travel of the arm. A roller attached to the pin on the front side of the front disk engages a stationary shroud when in the position shown, or may engage with the geneva gear when at the outer end of the slot. Locking and unlocking of the geneva gear is effected by engagement with a hollow sector projecting from the front face of the front disk.

The arm carrying the roller is normally held in its outer



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ASK FOR BULLETIN No. 838

GENERAL RADIO COMPANY GR

Cambridge 39, Mass.
NEW YORK · LOS ANGELES

position by a tension spring. A pivoted lever with a fulcrum, linkage and operating arm is normally held in the position shown in the figure, so that a cam face on the end of the lever forces the roller into the hollow sector as the disk is rotated. When the lever is pulled away from the sector by the action of an electromagnet the roller is allowed to engage the slot in the geneva gear.

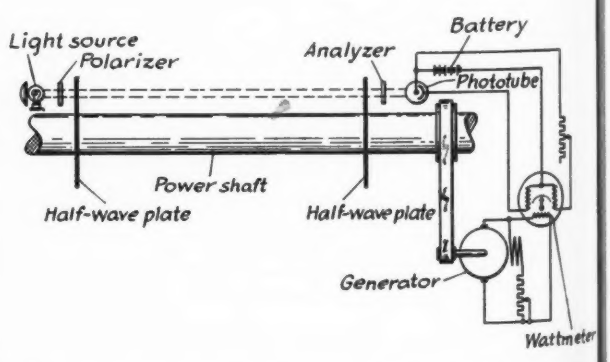
Cam Face Controls Operation

In operation the disk is continuously rotated in the clockwise direction, and the roller on the end of the arm tends to stay in its outer position through the combined action of the spring and centrifugal force. When the clutch is disengaged, however, and the roller is cammed into the hollow sector by the cam face, the driven geneva gear is locked in position by the roller riding upon the concave face of the shroud and also on the curved surface of the geneva gear even when released by the sector. When the clutch is to be engaged the magnet is energized, raising the lever so that the cam face no longer cams the roller into the hollow sector but allows it to engage a slot in the geneva gear, causing it to rotate counterclockwise as it is released by the sector. The roller is maintained in proper position by the convex face of the shroud and the geneva gear is advanced one tooth.

Phototube Measures Twist in Rotating Shaft

MEASUREMENT of torque or power transmitted through shafting is complicated by the difficulties inherent in determining the angle of twist in a rotating shaft. A new method employing polarized light and a phototube offers the advantages of requiring no slip rings or mechanical connections to the shaft and is claimed to be highly reliable. The device is covered by patent 2,313,-923, recently assigned to the Westinghouse Electric & Manufacturing Co.

As shown in the accompanying schematic diagram, the



Twist in shaft section between half-wave plates rotates plane of polarized light beam, affecting illumination of phototube and controlling current to wattmeter

twist in the power-transmitting shaft is measured by the angular displacement of two half-wave plates attached to the shaft some distance apart. Each half-wave plate is in the form of a circular disk and is made of a material such as cellophane which is doubly refractive. When

light enters a doubly refractive crystal in a direction no[t] parallel to the optical axis, the incident vibration is de-composed into two mutually perpendicular components which travel through the crystal with difficult velocities. If the thickness of the material is such that these com-ponents emerge out of phase by one-half wave length it is regarded as a half-wave plate for the particular light used.

Light from a monochromatic source is projected in a parallel beam through the polarizer where it is changed to plane polarized light before passing through the half-wave plates. When the optical axes of the two half-wave plates are displaced due to twist in the shaft the plane of polarization of the light emerging from the second plate is shifted by twice the angle of shaft twist. Thus the in-tensity of light emergency from the analyzer is a function of the twist in the shaft, and this fact is utilized by hav-ing the light fall on a phototube which controls the cur-rent flowing through an indicating meter.

Gives Direct Reading in Horsepower

In the arrangement shown a direct-current wattmeter is used, the indications being proportional to torque (as measured by current flowing through the phototube and stationary coils) multiplied by speed (as measured by cur-rent output of a generator driven by the shaft, flowing through the moving coil). The meter may thus be cali-brated in horsepower. Adjustment is made so that the light falling on the phototube at zero torque is one-half of the maximum illumination. A variable resistor is then set so that the two stationary field coils balance magneti-cally. As light falls on the phototube the current in the left-hand coil varies proportionately and becomes greater or less than the current in the right-hand coil. The residual magnetic field multiplied by the field of the mov-ing coil, which is proportional to the speed of the shaft gives the power.

Increased sensitivity may be achieved by providing multiple half-plate disks. Alternate disks are attached, respectively, to the shaft and to the free end of a long sleeve whose other end is secured to the shaft. Relative movement of the disks is equal to the twist in the shaft along the length of the sleeve.

Insures Torque Transmission to Driven Whee[ls]

CONVENTIONAL differential mechanisms, such as are used on automobile rear ends, divide the driv-ing torque equally between the two driven shafts regard-less of their respective speeds. This means that when one of the shafts experiences no resistance, as happens when a car wheel encounters a slippery road surface such as ice, no torque can be transmitted to the other shaft either. An alternative drive which permits different shaft speeds but insures torque transmission to at least one is covered by patent 2,315,299, assigned to C. F. Gobright Inc.

Essential parts of the mechanism are shown in the ac-companying illustration. The two driven shafts are actu-ated through jaw clutches which are capable of overrun-ning, though not at the same time. Driven members are splined to each shaft while the clutch driving mem-

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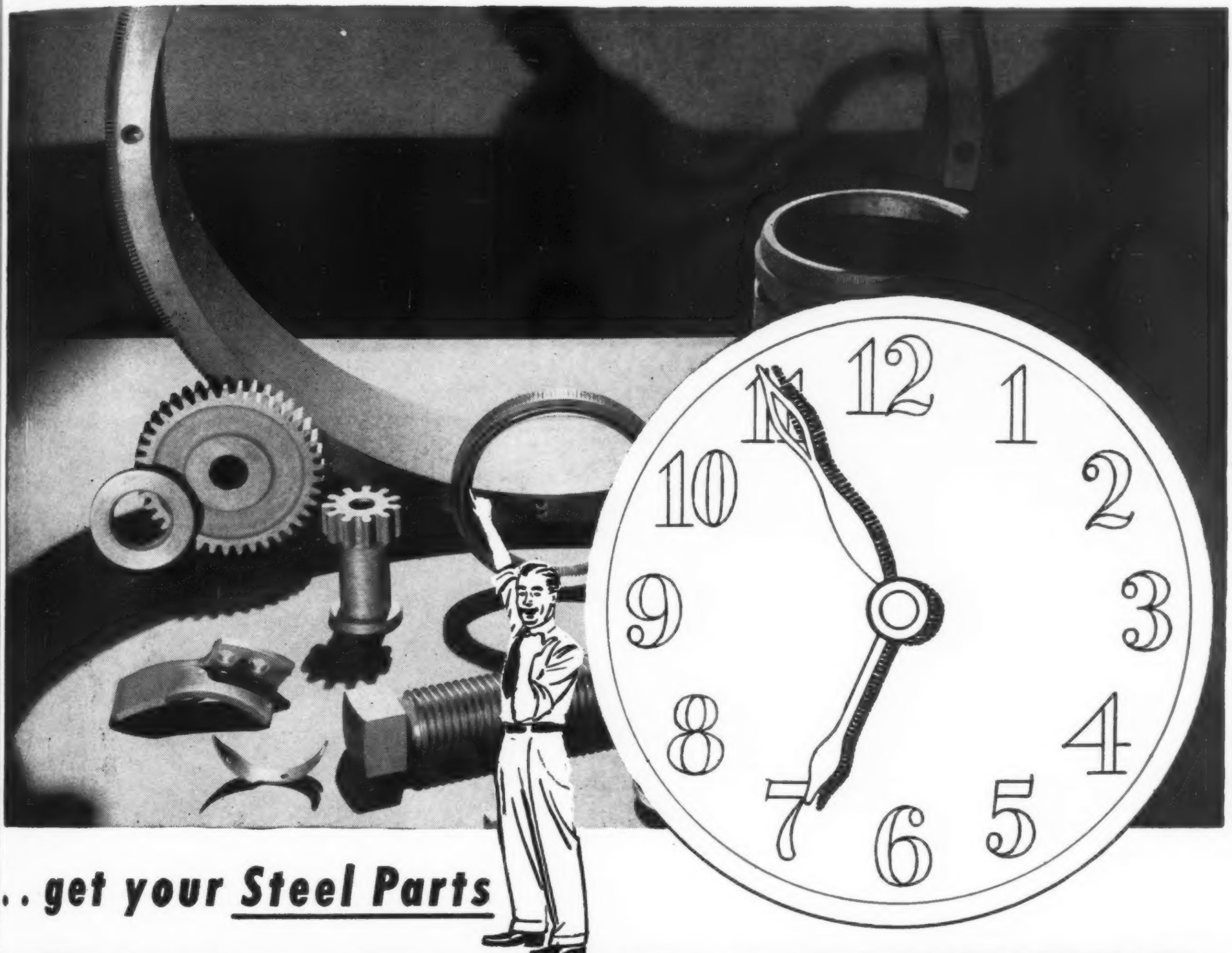

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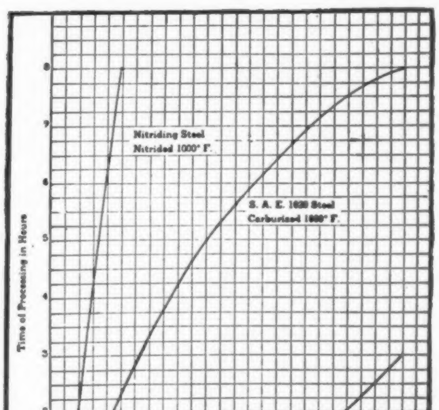
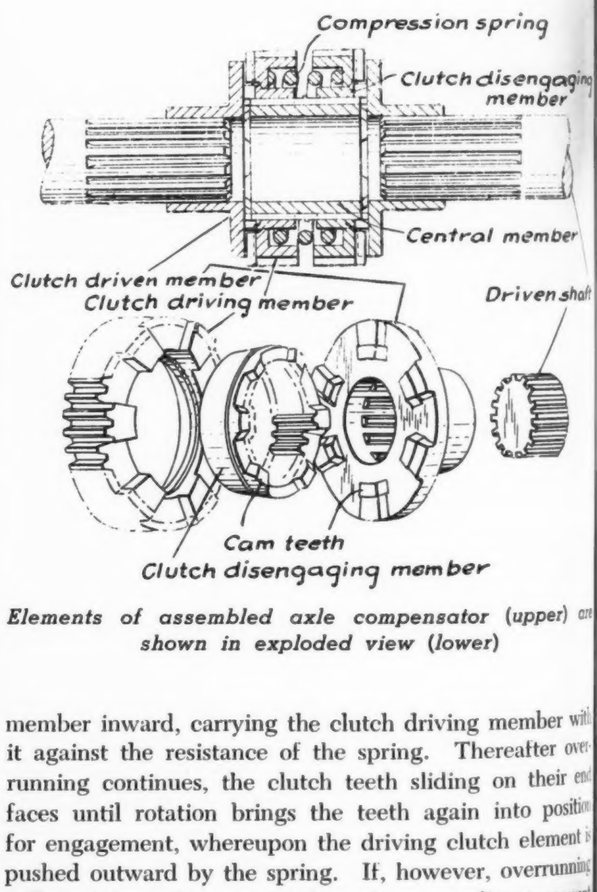


THE CHAPMAN VALVE MFG. CO.
INDIAN ORCHARD, MASS.

bers are splined on their outer circumferences for engagement with a housing attached to the main driving gear. The two clutch driving members are normally held in engagement with their mating driven elements by a helical compression spring, and the driven shafts rotate at equal speeds.

Overrunning Disengages Clutch

When conditions are such that their speeds should be different, as when a car turns a corner, the tendency is for the higher speed shaft to overrun the driven element. However, each clutch driven member is provided with a series of slanted cam teeth which mesh with similar teeth in the clutch disengaging member. The clutch disengaging members are both splined on the central member and have shoulders bearing against corresponding shoulders in the clutch driving members. Relative rotation between one of the clutch driven elements and the corresponding clutch disengaging member permitted by the backlash between the teeth, creates a pressure between the cam teeth which forces the clutch disengaging



Elements of assembled axle compensator (upper) are shown in exploded view (lower)

member inward, carrying the clutch driving member with it against the resistance of the spring. Thereafter overrunning continues, the clutch teeth sliding on their end faces until rotation brings the teeth again into position for engagement, whereupon the driving clutch element is pushed outward by the spring. If, however, overrunning still continues the cam teeth again cause disengagement and the process is repeated until the two shaft speeds become equal again.

It will be noted that as soon as overrunning occurs torque is transmitted only through the shaft which is running slower. Disengagement of both shafts simultaneously is avoided by proportioning the clutch driving members so that when one element is fully disengaged the two practically abut and the other element is prevented from moving out of engagement.

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Synthetic Adhesives

By Paul I. Smith; published by the Chemical Publishing Co. Inc., Brooklyn, N. Y.; 125 pages, 5½ by 8½ inches; clothbound; available through Machine Design, *$3.00 postpaid.*

As a practical aid in selecting the type of adhesive best suited to a particular purpose this book will be useful to the engineer concerned with joining different materials such as wood, metals, plastics, glass, rubber, textiles, paper, etc. Subjects covered in the ten chapters include: Dry gluefilm phenol-formaldehyde resin, phenol-formaldehyde liquid and solid glues, cellulose adhesives, polyvinyl cement, acrylic resin cements, chlorinated rubber and synthetic rubber cements, miscellaneous adhesives, use of synthetic adhesives in the manufacture of aeronautical improved plywood and high-density woods, and general applications of synthetic adhesives in industry. The final chapter on general applications is in the form of an extensive four-column series of tables listing various industries, applications, types of resins used for these applications, and main advantages of the selected resins.

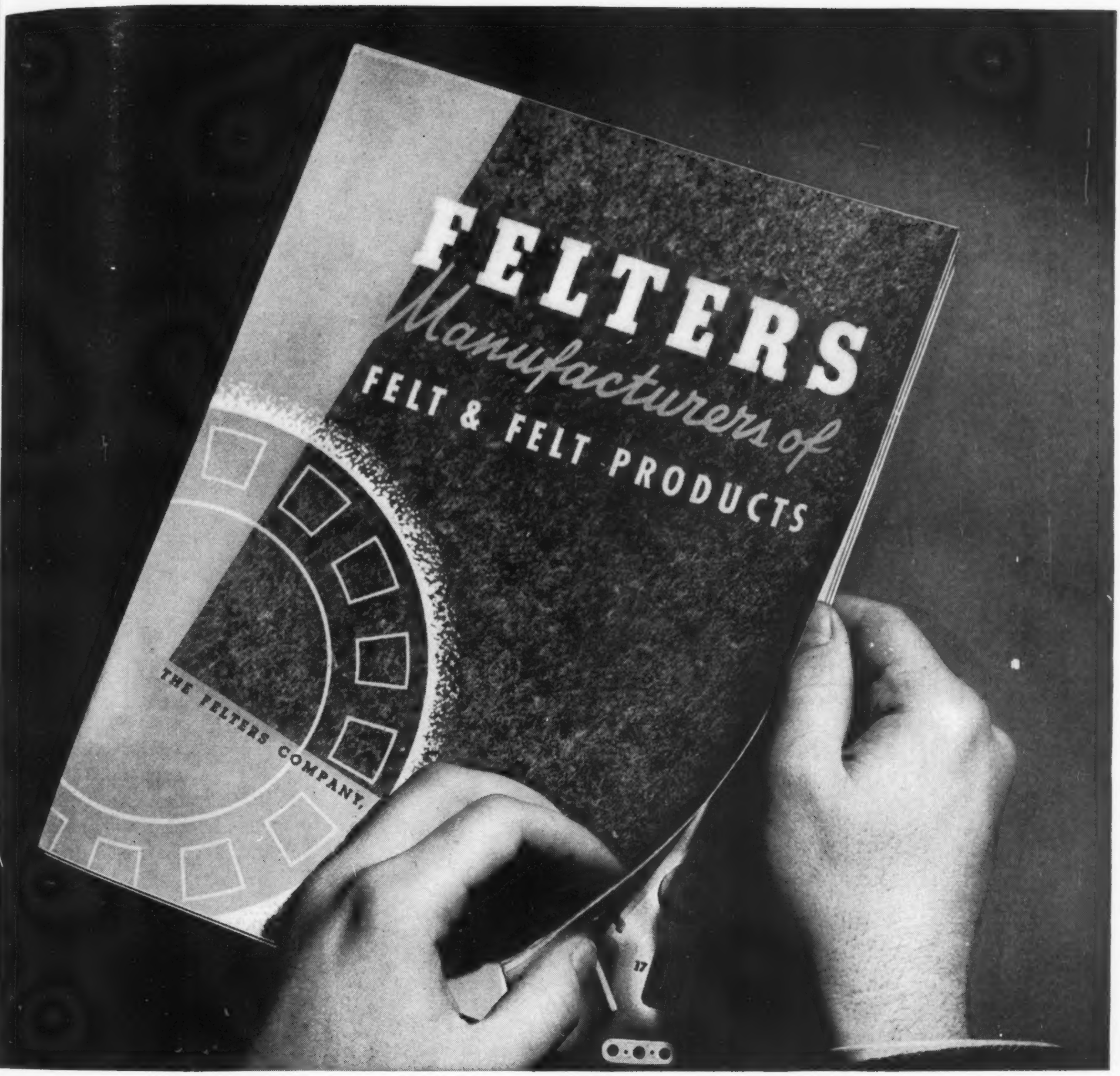
□ □ □

Dynamical Analogies

By Harry F. Olson, acoustical research director, RCA laboratories, Princeton, N. J.; published by D. Van Nostrand Co. Inc., New York; 196 pages, 5½ by 8½ inches, clothbound; available through Machine Design, *$2.75 postpaid.*

Affording a means of greatly simplifying the analysis of vibrating systems, the analogies between electrical, mechanical and acoustical systems are worthy of more consideration than hitherto has been accorded them by mechanical engineers. Certain basic elements in each system correspond with each other when reduced to mathematical form. For example, corresponding to the basic dimension of mass, analogous elements are inductance (electrical), mass (mechanical rectilineal), moment of inertia (mechanical rotational), and inertance (acoustical). Corresponding to the basic dimension of length are electrical charge, linear displacement, angular displacement, and volume displacement. Forces corresponding to mass times the rate of change of these basic dimensions are, respectively, electromotive force, force, torque, and pressure. From this starting point complete analogies are developed for all physical quantities in each of the four systems.

Purpose of the book is to enable the mechanical or acoustical engineer to set up the electrical network which is analogous to the problem to be solved, then to solve this electrical network by conventional circuit theory or even by experiment, and finally to convert the answer back into the original system. The reason for so doing is that at the present time electrical circuit theory has been



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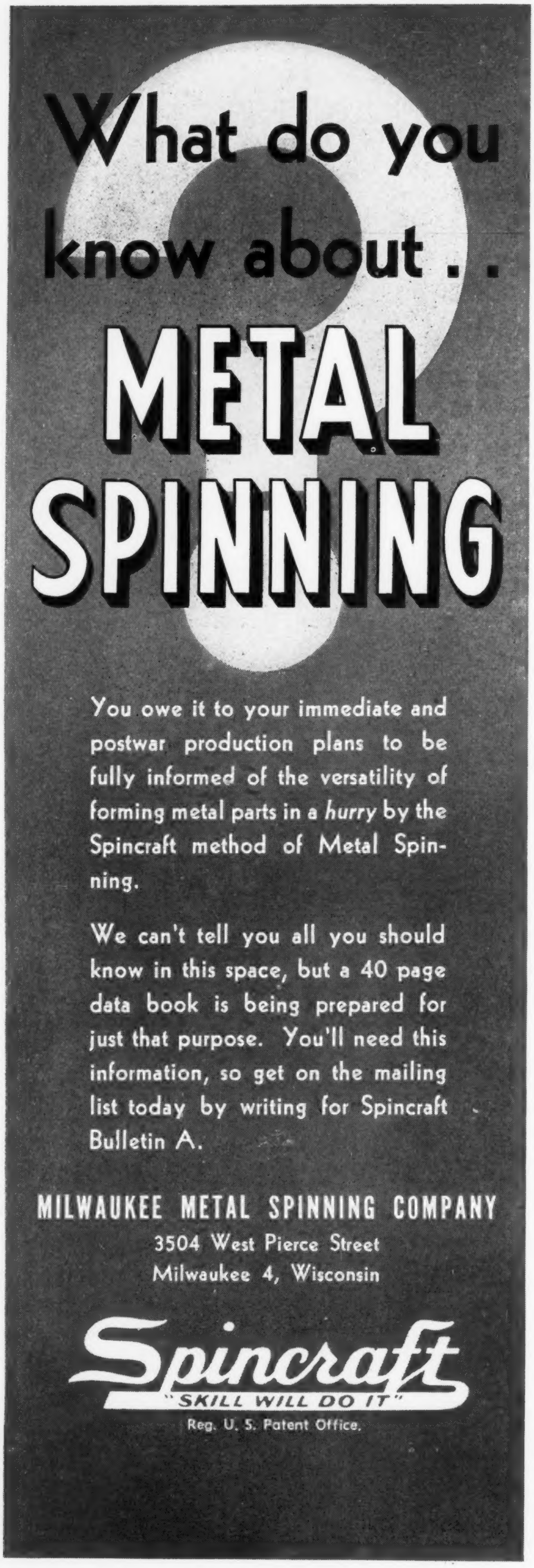

developed to a much higher state than the corresponding theory of mechanical systems.

Analogies discussed include vibrating systems of one, two and three degrees of freedom, corrective networks, wave filters, transients, driving systems (for converting electrical vibrations into mechanical or acoustical vibrations), generating systems (for converting mechanical or acoustical vibrations into electrical variations), theorems and applications. Among sample applications briefly discussed are an automobile muffler, an electric hair clipper with dynamic vibration absorber, a rotational vibration damper, various types of vibration isolation, and an automobile suspension system.

Inasmuch as mechanical engineers often are concerned with problems of cyclic heat flow, consideration of the thermal analogy would have added greatly to the value of the book. Perhaps the author will develop this additional analogy in a later edition.

□ □ □

Analytic Mechanics

By Sherman Daniel Chambers, Purdue University, and Virgil Moring Faires, Agricultural and Mechanical College of Texas; published by The Macmillan Co., New York; 375 *pages,* 6 *by* 9 *inches, clothbound; available through* MACHINE DESIGN, $3.75 *postpaid.*

Although a revision of Chambers' *Mechanics of Engineering,* this is a completely rewritten book. Explanations and discussions have been made more complete and a greater number of worked examples and problems included, resulting in a larger book although few new topics have been added.

The book is divided into twenty chapters, covering the usual elementary topics in statics and dynamics. With a large number of worked examples and no less than 1388 problems, many with answers, the book is well adapted to self study.

□ □ □

Resistance Welding Manual

Published by the Resistance Welder Manufacturers' association, Philadelphia; 286 *pages,* 6 *by* 9 *inches, semiflexible clothbound; available through* MACHINE DESIGN, $2.50 *postpaid.*

Covering exclusively all of the material on resistance welding which appears in the current edition of the *Welding Handbook,* this manual is a practical reference which treats all aspects of the subject. The six chapters, each prepared by a committee of authors and consultants, deal with the fundamentals of resistance welding, resistance welding processes, resistance welding machines, controls and timing devices, electrodes, electrode holders and conductors, definitions and symbols.

Although much of the discussion is highly specialized, having to do with details of the processes and of the machines used, designers considering methods of fabrication will find some useful information on the advantages and limitations of the various resistance welding processes for different kinds of jobs and materials.

DESIGN ABSTRACTS

Cooling Problems in Tank Engines

ONE of the major obstacles to the use of liquid-cooled engines in combat vehicles is the greater airflow required and consequent difficulty involved in their cooling. The reason for this is that whereas the fins of air-cooled engine cylinders in operation will rarely drop below the perfectly satisfactory temperature of 350 degrees Fahr., 225 degrees is usually considered the practical limit for radiator temperature in the cooling system of a liquid-cooled engine. Assuming operation at an ambient temperature of 125 degrees, the water-cooled engine permits only a 100-degree temperature differential while the air-cooled engine has a temperature differential of 225 degrees in which to work.

Efficient Baffling Is Requirement

On the other hand, successful cooling of finned cylinders requires both an adequate flow of cooling air and efficient baffling of the individual cylinders. Much study has been devoted to baffling of finned cylinders, notably by the N.A.C.A., and this feature may be considered no longer subject to radical change. Fan capacity may be adequate but it is often found that proper cooling is still not realized because inlet and outlet passages are so tortuous and obstructed as to reduce the airflow below the critical value. Unless care is taken to avoid air recirculation, cooling efficiency may be seriously reduced thereby. Too, high fuel temperatures may reduce vapor lock and require auxiliary equipment to overcome that effect.

In any type of combat vehicle, particularly with tanks the difficulties of cooling are tremendous because the engine virtually is enclosed, especially in action. Either air cooling or water cooling, properly designed, is satisfactory, except that certain difficult military situations likely to arise point to air cooling as being the more satisfactory of the two.

Cooling System Faces Hazards

Leakage, puncture of the radiator by enemy fire, clogging of radiators from without by dust, dirt and twigs or restrictions within the system for the same reasons are real problems. One of the most urgent current demands from the field is for steam jennies to clean out radiator cores. In many theaters of operations water is neither available nor suitable for use, aside from the ability of the crew to replenish water supply in combat. Maintenance difficulties due to vibration, and other factors, all militate against the use of water-cooled engines. The problem of liquid-cooled engines under winter condi-

tions likewise is a serious one. A letter I recently received from an Ordnance officer in Russia said, "If people could see the difficulties involved in handling liquid-cooled tank engines in a Russian winter, the argument in favor of air cooling would be settled once and for all."

The additional weight involved in water cooling is another argument against its use. This is a major item affecting the characteristics of both mobility and protection. The Ordnance department would prefer a ton of weight in additional armor to a ton of dead engine weight. Differently expressed, for equal horsepower and performance characteristics, an engine weighing one ton more than another automatically costs us one ton of protection if gross weights are comparable.—*From a paper by Lt.-Col. R. J. Icks, Ordnance department, presented at the recent S.A.E. War Materiel meeting in Detroit.*

Thermoplastic Has High Heat Resistance

ARTICLES molded from a new high heat-resistant methyl methacrylate resin molding powder will not soften appreciably or distort when exposed to a temperature of 212 degrees Fahr. This is 30 to 40 degrees above the useful temperatures for similar articles made from other commercial thermoplastic molding powders. Airplane flying light lenses, dial and meter faces, medical and dental instruments, and airport and railroad signal light lenses are among the applications for which this new Lucite plastic molding powder has been developed.

The molding powders are prepared from the same basic materials as the cast sheets, quantities of which now are formed into noses, gun turrets, navigation blisters and other transparent enclosures on military planes. The high heat-resistant formula has approximately the same mechanical, optical, electrical and molding properties as the general-purpose molding powders now used for reflectors on military vehicles, army compasses, navy control equipment indicators and other items of ordnance. Articles made from it may be clear, or the powder may be dyed or pigmented to desired colors.

Can Be Used with Existing Equipment

The new formula has been developed for use in existing compression, injection and extrusion equipment. Best technique for molding requires injection temperatures 30-50 degrees Fahr. higher than are used for regular Lucite. However, it is advisable to employ the lowest temperature at which the die cavities will fill.

The new powder will be available also in granular form for compression molding, and has all the temperature characteristics of the injection or extrusion powder. Good extrusion results are obtained by using a short screw and a low screw speed, preferably five revolutions per minute or less. Stock must be thoroughly dried. A moisture content not exceeding .02 per cent is necessary for best extrusion results.— *From a talk by G. M. Kuettel, Plastics Department, E. I. duPont de Nemours & Co., at the recent annual meeting of the Society of the Plastics Industry in Chicago.*

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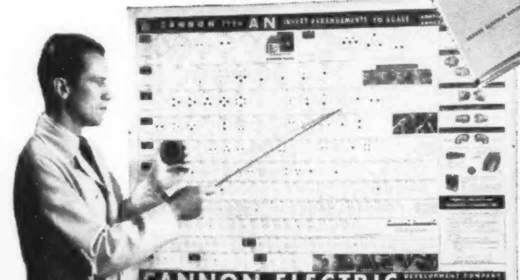

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Short-Circuit Sammy is a genius at getting into trouble, but he raises some pretty good questions in the Cannon Plug training film. His pedagogical characteristics are confined to showing how not to do certain operations

Designing Ring Springs

(*Concluded from Page* 127)

The tension stress σ_t in the outer ring at $P_1 = 100{,}000$ pounds is obtained from Equation 7,

$$\sigma_t = \frac{100000 \times .25}{\pi \times .584 \times .095} = 143{,}000 \text{ psi}$$

Since $A_o = A_i$ in this case, from Equation 5, the foregoing will also be equal to the compression stress in the inner ring. Usually in practice, however, the inner ring area A_i is made smaller than A_o since it has been found from experience that higher working stresses may be used in compression than in tension. For example, draft gear springs have been designed for a circumferential tension stress of 125,000 pounds per square inch in the outer rings as compared with a compression stress of 210,000 pounds per square inch in the inner rings when the spring is solid (2).

Assuming that the design proportions of the spring are so chosen that the projected contact length b (*Fig.* 3), is .79-inch at a load of 100,000 pounds, then from Equation 11 the compressive stress σ_c' in the contact area is

$$\sigma_c' = \frac{143000 \times .584}{2 \times 4.58 \times .79 \times .97} = 11{,}900 \text{ psi}$$

Adding this to the tension stress $\sigma_t = 143{,}000$ pounds per square inch, the equivalent stress in the outer ring becomes $143{,}000 + 11900 = 155{,}000$ pounds per square inch. This is a slightly higher value than would be obtained if the contact compressive stresses were neglected.

As an approximate indication of loads and deflections possible for this type of spring the accompanying table, Page 127, published by Edgewater Steel Co., is useful.

Ring springs must be provided with lubrication since a dry spring will gall and stick under comparatively light service. A graphite grease may be used for this purpose. Such a lubricant not only decreases the friction coefficient, thereby reducing the heat generated, but also acts as a coolant.

In general, ring springs must also be provided with guides, either internal or external, to prevent sidewise buckling. Such guides are usually provided with a clearance of about two per cent of the diameter.

Besides the applications previously mentioned, ring springs have been used in a variety of types of service including elastic supports for bedplates in presses, shock absorbers for guns, flexible draw-bars for trailers and buffers for cranes.

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SUPERCHARGERS

(*Continued from Page* 139)

and the impulses caused by the passage of impeller blades past the diffuser vanes. The energy and destructive power of these vibrations is sometimes surprising, and require special care and experience to determine the necessary massiveness and structure to prevent premature fatigue failures.

For aerodynamic reasons—to preserve the proper proportions of the flow passages—the capacity of a given type of supercharger should not be increased faster than the square of the characteristic dimension, which is usually the diameter, $Q \propto D^2$. In any series of similar machines the weight increases with the cube of the characteristic dimension, $W \propto D^3$, hence $W \propto Q^{3/2}$, indicating that the larger-capacity machines tend to become relatively heavy. This aspect of the problem becomes so serious that there is a strong incentive to cramp and compromise the design in order to maintain more nearly proportionate weights as the horsepower rating is increased. For this reason the impression is sometimes given that the art is not progressing, since real gains in basic merit are required to stand still with respect to efficiency while constantly progressing in capacity and altitude rating.

Gear-Driven Supercharger Is Standard

The commonest type of supercharger, embodied in practically every major commercial and military aircraft engine today, is the radial centrifugal, gear-driven from the rear end of the engine crankshaft. The speed is usually stepped up through two sets of gears to give impeller speeds of from six to ten times crankshaft speed.

For conservative performance at moderate altitudes, as in typical airline service, a single-stage, single-speed drive is adequate, but for high-performance military planes it has been found expedient to employ some form of multi-speed drive for the following reasons:

If a centrifugal supercharger is operated at constant speed the pressure ratio and volumetric capacity are substantially independent of the initial pressure and density of the air. But the power is proportional to the weight flow, and hence to the density and, of course, the final pressure increases in proportion with the initial pressure. A supercharger designed to give a full-power manifold pressure of 45 inches Hg at 25,000 feet, where the pressure is 11.1 inches and the density is .0343 pounds per cubic foot, would therefore give a manifold pressure of 121 inches at sea level and the power required to drive it would be 2.23 times normal. To avoid wrecking the engine immediately the throttle would have to be closed to drop the M.A.P. from 121 to 45 inches. But the supercharger would still extract an unnecessarily large block of power from the engine, and the heat of compression—starting with a relatively high initial air temperature—would tend to cause heavy detonation unless the M.A.P. and engine power were reduced still further. In order to maintain the maximum net engine power or economy at low altitudes it is therefore necessary to regulate the supercharger speed so that no more than the required pressure is generated at each level. In fact, it is one of

the axioms of good flight engineering that the throttle should be kept open as far as practicable under all operating conditions.

Gear-driven superchargers, either single or two-stage, are now frequently provided with a two-speed drive having a mechanical clutch for shifting from low to high gear ratio. More recently, fluid couplings have been introduced, particularly in German engines, to provide continuously variable speed control. An excellent description of the details of these drives is contained in a recent article by F. M. Kincaid of the Wright Aeronautical corporation(7). All of them are characterized by a relatively high degree of mechanical complexity, especially when fully automatic control is provided to maintain the desired M.A.P. independently of altitude.

Turbine Drive Permits Speed Control

The exhaust gas-driven turbosupercharger represents a practical solution to the drive problem in terms of mechanical simplicity, flexibility and power economy. As may be seen from the schematic diagram of *Fig.* 5, the impeller of the turbosupercharger is direct-connected to a single-stage turbine driven by the engine exhaust gas. The speed is controlled in a simple manner by allowing excess gas, not required for turbine operation, to escape through the wastegate instead of through the turbine nozzles and wheel. This makes it possible to hold constant manifold pressure, with no throttling and no extraction of engine crankshaft power, from sea level to critical altitude. No more than the minimum required power is drawn from the turbine at any time, so that the excess back-pressure which it imposes upon the engine is likewise never more than sufficient to supply the desired manifold air pressure. The significant fact here is that the reduction in engine power caused by this increased back-pressure is usually much less than the crankshaft power required to drive an equivalent geared supercharger(8).

As indicated in *Fig.* 5, it is common practice to employ an internal geared supercharger in conjunction with the turbo, with an intermediate intercooler. With a relatively low pressure ratio the internal stage has no speed control, since it will not generate excessive manifold pressures with open throttle at sea level if the turbo is cut out.

Views of the General Electric turbosupercharger of the type used in the Boeing Flying Fortress and Consolidated Liberator bombers and in the Lockheed Lightning and Republic Thunderbolt pursuits are shown in *Figs.* 6 and 7. Although military necessity prevents the disclosure of complete technical details, the major features may be indicated as follows:

Referring to the cutaway view, *Fig.* 6, the engine exhaust stack connects to the nozzlebox *A*, which is directly above the turbine wheel *B*. The hot exhaust gases expand through the nozzles *C* and impinge upon the turbine buckets *D*, as shown in *Fig.* 5. The pressure in the nozzlebox is regulated by means of the waste gate *E*, which vents excess gas to the atmosphere.

Power developed is transmitted through the shaft *F* to the impeller *G*, which is also shown in *Fig.* 5. The diffuser passages *H* discharge into a scroll-type collector in the outer portion of the compressor casing *J*, which has a radial outlet. The rotor assembly is carried in a ball bear-

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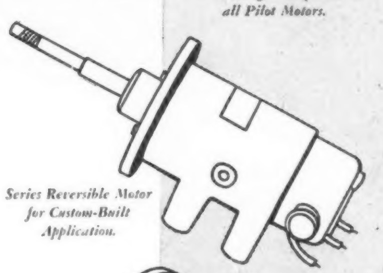

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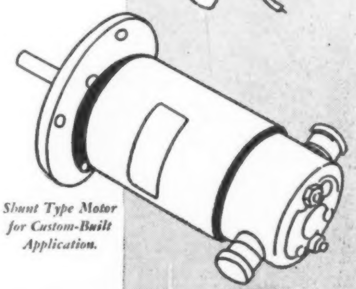

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ing *K* at the impeller end to take the thrust load, and a roller bearing *L* at the turbine end to allow for expansion of the shaft.

The baffle ring *M*, also shown in *Fig.* 7, serves to shield the compressor casing from the heat of the nozzlebox, and its inner portion supports the nozzle ring.

The lubricating oil pump *N* is driven by a worm gear from a worm sleeve keyed to the shaft. The pump is of the gear type, and actually consists of two separate pumps within a single casing. One element supplies oil under pressure to the gears and bearings. The other element is a scavenging pump which removes oil from the lower part of the bearing housing *P* and returns it to the supply tank.

The impeller is machined out of an aluminum alloy forging. The compressor casing, diffuser and bearing are aluminum alloy castings. Turbine parts are made of alloy steels especially developed to withstand the high stresses, elevated temperatures and corrosive action of the exhaust gases. Data on various materials of this latter class are given in Reference(9).

As indicated in *Fig.* 5, regulation of the turbine waste gate is effected by means of a hydraulic servomotor containing a pressure-sensitive element connected to the engine exhaust stack ahead of the turbine. By adjusting the boost control lever, the pilot sets the regulator to hold a constant nozzle-box pressure which will maintain the desired engine intake manifold pressure.

Waste Heat Is Utilized

The underlying thermodynamic principles of the gas turbine drive are essentially the same as those applying to the conventional single-stage impulse-type steam turbine. The success of the turbo-supercharger is based upon the development of an adequate mechanical design to enable the turbine to utilize the energy of the high-temperature engine exhaust gases with satisfactory reliability and life and without undue penalties in weight or size.

The available energy, in Btu per pound, of engine exhaust gas for various temperatures, pressure ratios and fuel-air ratios is given in *Fig.* 8. Data are based on recent information on the specific heat of gases reported by Heck(24) and Ellenwood, Kulik & Gay(25), and are believed to be considerably more accurate for this purpose than the energy values based on the properties of ordinary air. (To convert the available energy to horsepower per pound of gas per minute, divide Btu per pound by 42.4). It should be noted that this is primarily *thermal* energy representing the waste heat of the exhaust gases and that the pressure ratio required to release a given amount of power decreases rapidly as the initial gas temperature increases.

Actual exhaust gas temperatures, in the case of high-powered military engines, vary from about 1200 to 1700 degrees Fahr., the higher figures being associated with higher outputs and leaner fuel-air ratios. Assuming a representative temperature of 1500 degrees Fahr., it is of interest to determine the available energy per pound of gas for a presure ratio of 2.69, which corresponds to sea-level back pressure ($P_1 = 29.92$ inches) in the engine exhaust stack at 25,000 feet ($P_2 = 11.1$ inches). The conservative "cruising lean" value of .075 will be used for

the fuel-air ratio although richer mixtures would be used for full power.

From *Fig.* 8, for $p_1/p_2 = 2.69$ and $T_1 = 1500$ the available energy is 122 Btu per pound, or 2.88 horsepower per pound per minute. This figure must be multiplied by the turbine efficiency to obtain the actual shaft output.

The engine requires slightly less than a pound of air per pound of exhaust gas. It will be assumed that the turbocompressor delivers this air at sea-level pressure at the carburetor, which will require a pressure $p_2 = 31.67$ at the entrance to the intercooler, with a compressor efficiency of 70 per cent. From Equation 2 and *Fig.* 4 the required power input to the compressor will be $(.00573 \times 148)/.7 = 1.21$ horsepower per pound per minute.

This means that the turbosupercharger can maintain sea-level pressures at both ends of the engine at 25,000 feet so long as the turbine efficiency is greater than $1.21/2.88 = 42$ per cent. This is entirely practicable, even after allowing for losses due to gas leakage, etc., and is the basis of most turbo applications.

Available and Required Powers Balance

As the altitude increases, the compressor must develop a greater pressure ratio, which requires more power. But with constant engine inlet and exhaust pressures, the pressure ratio across the turbine also increases with altitude, so that the power available continues to balance the power required. The ultimate level above which this balance can no longer be maintained is determined either by the attainment of the maximum allowable turbo speed or else by loss of turbine efficiency at excessively high pressure ratios. Beyond this critical altitude the carburetor pressure and engine power begin to decline in the normal manner.

From the energy standpoint it might be said that the turbo thrives on high exhaust gas temperatures, as indicated by the curves of *Fig.* 8. However, at temperatures in the vicinity of 1700 degrees Fahr. the cooling problem becomes serious to such an extent that it has been thought in many quarters, particularly abroad, that turbine operation at such temperatures was impracticable. The fact is that the ability of the turbine materials to withstand these high temperatures is due largely to the fact that no parts of the machine actually reach the full temperature of the gas. Practically all parts are cooled by intense radiation to the surroundings and the nozzlebox is cooled about 400 degrees below the gas temperature by the additional effect of a stream of "rammed" air. The effective temperature of the gas impinging upon the buckets is reduced several hundred degrees by virtue of the conversion of thermal energy into kinetic energy in the high velocity jets(12). It appears, therefore, that present engine exhaust temperatures are nearly at the optimum level for turbo operation—if they were much higher the necessary reduction in turbine speed or life would be serious, and if they were much lower the performance would be impaired by the reduction in available power.

The problem of "afterburning" may be mentioned in this connection. Since aircraft engines are always operated with rich fuel-air mixtures at high powers, the exhaust gases contain considerable quantities of carbon monoxide and hydrogen; up to 14 per cent CO and 7 per cent H_2 (13). These highly flammable gases will burst

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into flame if they come into contact with air before cooling below their ignition temperature (roughly 1100–1200 degrees Fahr.) Because of their visibility at night these flames must be suppressed in the case of bombers, many of which are provided with bulky quenching devices for this purpose. In turbosupercharger installations afterburning can cause local over-heating if the gases come in contact with sufficient air, by entrainment or otherwise, before passing completely through and away from the turbine. But with suitable precautions to avoid this, the turbo has a salutary effect in helping to suppress visible flames by cooling the gas several hundred degrees due to the conversion of heat into work.

It is extremely difficult to make valid comparisons of the relative performance of a given airplane with gear-driven or with turbosupercharger for the simple reason that, for optimum results, the plane must be designed specifically for the particular installation used. This is especially true in pursuit ships, where the supercharger has a marked effect upon fuselage shape and weight distribution. Moreover, the requirements of a light "hedge-hopping" plane are apt to be quite different from those of a heavy, high-altitude design. The performance of any supercharger, and particularly the turbo, depends so greatly upon the excellence of the installation that it is entirely possible to reverse the results of any such comparison by suitably butchering the job. It is significant, however, that the gear-driven superchargers have had the benefit of much more intensive work on the installation problem, so that some account should be taken of the potential as well as the actual merit of the turbo. Even the latest and best turbo installations are not as closely and completely integrated into the power plant as they might be. But the fact remains that the turbo-supercharger enjoys an initial and basic thermodynamic advantage which becomes even more marked at higher altitudes. And finally, it is still more significant that four of the outstanding planes that this country has contributed in the present war, the B-17, B-24, P-38, and P-47 are turbo-equipped.

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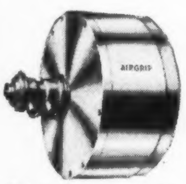

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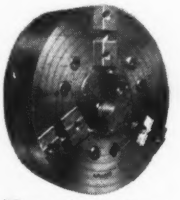

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Wartime Metallurgy

(*Continued from Page* 150)

which then precipitates the carbides in a finely divided form. It is also probable that the immediate surface is altered to martensite. The austenitic core insures good impact resistance. Nickel, chrome and other alloying elements are sometimes added. These steels are excellent for rail sections subject to unusual service such as crossing and switch frogs; for hoisting rope sheaves and drum laggings; for excavating dipper teeth, etc.

SILICON: Silicon forms no carbides, dissolving fully in the ferrite. In fact, it has in a way an opposite effect, promoting a breakdown of the carbides or "graphitization". Steels high in silicon must, for this reason, be heated in controlled atmosphere to prevent decarburization, and will not readily carburize. Its contribution to hardenability and grain control is not large. It shows good resistance to tempering. Little use is made of the plain silicon steels except in the electric grades of 2½ to 4 per cent Si with very low metalloids.

The early "silicon structural" steel was misnamed, carrying as it did only .20 to .30 per cent Si, its strength being primarily obtained by running the Mn up to .70 per cent to .90 per cent and the carbon to .30 to .40 per cent. Silicon is used, however, in many later low-alloy structural steels in amounts up to 1 per cent in which it serves as a ferrite strengthener.

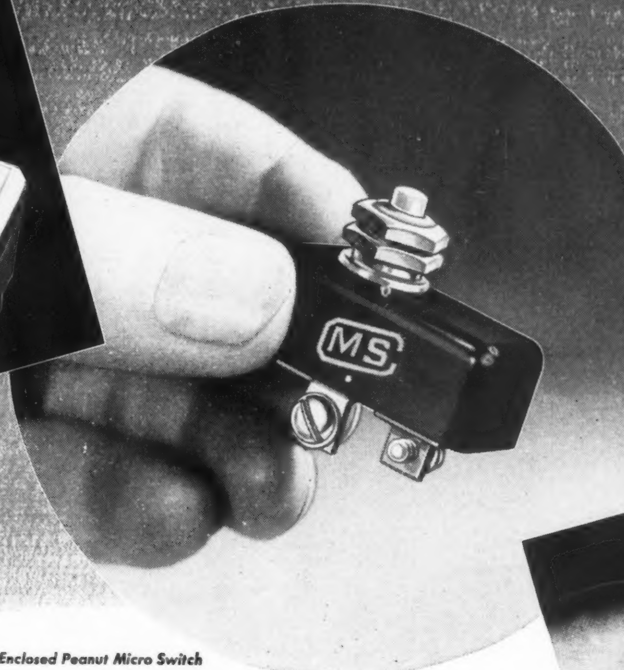
Of considerable importance are the silico-manganese steels of which SAE 9260 is typical; silicon running 2 per cent, manganese a little under 1 per cent. The presence of the silicon seems to contribute good wear qualities, and the alloy exhibits an excellent combination of strength and toughness. Silicon also contributes to good surfaces in the as-rolled condition in which a good deal of this steel is used for leaf springs.

Tool steel grades, to which some one-quarter per cent chrome and an equal amount of vanadium have been added (this last eliminated for the duration) have found wide applicability for punches, chisels and other shock-loaded tools. The chrome improves hardenability somewhat and contributes further to wear qualities. Grades are also in use with .5 per cent Mo substituted for the Cr with still further improvement in hardenability. All of these grades form admirable and economical steels and have been widely and successfully applied by the writers' company to such applications as small and heavily loaded pinions, particularly where abrasion is present, shear blades, heavily stressed trunnion pins, punches, ratchets, cams, etc. These steels require controlled atmosphere for satisfactory heat treating.

Silicon, with higher carbon, is also used in the graphitic steels for drawing dies, the Ford crankshaft alloy, and other applications (36) and (37).

COPPER: Copper is added in amounts up to about one-half per cent to nearly all the low-alloy structural steels, and to many others, notably copper-bearing sheets, to impart atmosphere corrosion resistance. It apparently forms a tight, adherent oxide which hinders further oxidation. In the amount used it does not influence materially the weldability, workability, or other properties.

It has also been added in amounts as high as 3 per cent for its precipitation hardening effect, as in the Ford



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Skeleton Peanut Micro Switch

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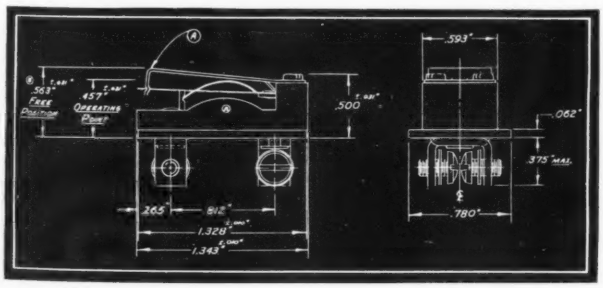
Designed for post war use, the Peanut Micro Switch so filled the bill for a number of military requirements that we are now making them in large scale production.

This diminutive switch is smaller than the basic Micro Switch, more robust, more resistant to vibration, is lighter, lower in cost and has a higher electrical rating.

The Peanut Micro Switch is so designed that it can be used without an enclosing case, allowing the actuating movement to be applied anywhere over a large portion of the upper spring. Used with a Bakelite case, a convenient stem mounting is provided. The construction allows inherent overtravel beyond the point of operation sufficient for most uses.



The unenclosed Peanut Micro Switch weighs .019 lb., encased in Bakelite housing the weight is .048 lb. This switch resists vibration and acceleration up to 300 times gravity. Operating force is 32 ounces maximum and the movement differential is .036" maximum.

Contact separation up to .085" can be varied in manufacture to meet requirements. This wide gap is particularly valuable on DC loads. To provide the high unit contact pressure for low voltage applications, contacts of 99.95% pure silver are formed with a knurled surface.

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Micro Switch Corporation, Freeport, Illinois

Branches: 43 E. Ohio St., Chicago (11) • 11 Park Place, New York City (7)
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Peanut Micro Switch Ratings

DIRECT CURRENT

	HEATERS		LAMPS		MOTORS—RELAYS & SOLENOIDS	
VOLTS D.C.	Sea level amp.	45,000 ft. amp.	Sea level amp.	45,000 ft. amp.	Sea level amp.	45,000 ft. amp.
†) 24-28	40	40	7	7	30	25
*) 110-115	1.5	1.2	1.5	1.2	1.2	0.75
*) 220-230	0.6	0.6	0.6	0.6	0.5	0.3

†) Aircraft use of 25,000 operations **) Not less than 100,000 operations*

ALTERNATING CURRENT

Volts A.C. 60 cycle	Heater amp.	Lamp amp.	Inductive 75% P. F. Amperes
110	20	10	15
220	10	5	8

Not less than 100,000 operations

MOTORS 110-220 V. A.C. motor rating ¼ h.p.

The trademark MICRO SWITCH is our property and identifies switches made by Micro Switch Corporation

MICRO SWITCH

Made Only By Micro Switch Corporation . . . Freeport, Illinois

graphitic crankshaft alloy previously mentioned. While copper is not a carbide former it will dissolve in the ferrite in equilibrium, only up to three-quarters per cent, and it is to this that it owes its precipitation hardening effect.

SULPHUR: Used to impart free machining properties, sulphur has been discussed in Part IX*. The high manganese grades (A1S1-C1112) for carburizing and X-1335 (A1S1-C1137) for through hardening, are recommended for stressed parts to avoid danger from sulphur segregation and to give some depth hardening.

PHOSPHORUS: This was also discussed in Part IX. It is employed in high-strength structural steels, generally in combination with other elements, to improve the as-rolled physicals.

LEAD: Lead has been added, generally in amounts under one-quarter per cent, to carbon and alloy steels to improve machinability. It is insoluble in the steel and must be introduced in such a fashion as to be in a finely divided state. It is reported as having no appreciable effect on the physicals or heat treatment.

TUNGSTEN: This alloying element acts in steels much the same as molybdenum, except that double the quantity is required for the same effect. While it has been employed in constructional steels it is too expensive an addition for regular use, and indeed there is little point in borrowing from its use in tool steels.

ALUMINUM: Additions of aluminum for grain size control have been extensively discussed in Part VI, and for nitriding in Part XI.

OTHER ALLOYING ELEMENTS: A variety of other special alloying additions have been experimented with in recent years, such as titanium, zirconium, boron and calcium, employed singly and in combination, giving rather amazing results by improving hardenability and other qualities with extremely small additions (19). The effect seems to be more of enhancing the effect of other alloys, rather than of any intrinsic value they may have, although this has been by no means definitely established (35). Boron, in particular, (25) and (26) has been used recently with good results. Titanium has been extensively investigated by Comstock, (27) and (28). He reports a reduction in dendritic structure, and an ultra-high grain-coarsening temperature. Much more will probably be heard in the future of these new additions of the family of alloying elements.

National Emergency Steels

Shortages of alloys brought about by the war effort have forced a scrutiny of the commonly used analyses to the end that strategic elements be used more economically. This series of articles has brought out the fact that many applications of high and medium alloys could as well be served by lower alloying contents. A survey of scrap return to mills showed that residues, with occasionally small additions, could serve many applications and the NE series of steels were accordingly set up. There is not space here for an exhaustive discussion of all the analyses that have been tried and indeed it would be needless duplication as they have been well covered elsewhere. Suggested references include (29), (30), (31)

*MACHINE DESIGN, April, 1943.

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and (32), as well as excellent handbooks (17) and (18).

Considerable difficulty has been experienced in predicting the analyses of heats. As a result, most of the steels have gone to large users rather than to warehouse stocks. Recent practice, however, appears to have centered for this latter purpose on those grades embodying a "little bit of everything", as in the nickel-chrome-moly type NE 8600, and on those with manganese and silicon, the NE 9400 type. The through-hardening grades, NE 8442 and 9442 have been covered in the table and in *Fig.* 108. While admirable steels for many purposes, they really afford too high a hardenability for some applications. The situation is changing rapidly, making it difficult to state what the condition will be in future.

In the writers' opinion it is regrettable that the NE steels were not drawn up around hardenability and grain size requirements, rather than analysis. Such a practice would permit full flexibility to the mills, with the maximum of efficient employment of our alloying resources. It is difficult to understand how, in the face of modern knowledge of the functions of the alloying elements, it can be maintained that it is necessary to know the exact chemicals on 90 per cent of the applications, even with today's severe war requirements. Such a practice would greatly facilitate warehouse stocking, a neglect of which has contributed today to shortages of many components. NE steels should be reviewed and reconstructed with the aid of the most advanced metallurgical knowledge.

(End of Series)

REFERENCES

(For References 1 to 16, other than those listed below, see previous article, July, 1943)

1. Bain, Edgar C.—*Functions of the Alloying Elements in Steel*, American Society for Metals, 1939.
3. Bullens, D. K.—*Steel and Its Heat Treatment*—Vol. II, John Wiley & Sons, 1939.
14. *Metals Handbook*, A.S.M., 1939.
15. Boston, O. W.—"Machinability of National Emergency Alloy Steels", *Metal Progress*, April, 1943.
17. *NE Steel Handbook*, Compiled from contributions to *Steel*, 1943.
18. *National Emergency Steels*, Compiled from various contributors by A.S.M., April, 1943.
19. "Report on Special Alloy Addition Agents"—A.I.S.I., Nov., 1942.
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21. Jameson, Alfred S.—"Chromium Steel for Ball Bearings; Inspection and Heat Treatment", *Metal Progress*, May, 1941.
22. Patton, Walter G.—"Moly Steels Combine High Physicals, Workability", MACHINE DESIGN, Jan., 1941.
23. Zimmerli—"Effect of Temperature on Coiled Steel Springs Under Various Loadings", *Transactions*, A.S.M.E., May, 1941.
24. Wahl, A. M.—"Which Spring Material?", MACHINE DESIGN, Nov., 1942.
25. Tisdale, Norman F.—"Boron—A New Vitamin for Steel and Cast Iron", *Metal Progress*, March, 1942.
26. Jameson, A. S.—"Effect of Boron Containing Additions on NE Steel", *Iron Age*, March 25, 1943.
27. Comstock, George F.—"The Effect of Titanium . . .", *Transactions*, A.S.M., Sept., 1940.
28. Comstock, George F.—"Some Properties of Phosphorus-Titanium Steels", *Transactions*, A.S.M., June, 1942.
29. Mitchell, John—"Relative Effect of Elements on Alloy Steels", *Metal Progress*, July, 1942.
30. Wellauer, E. J.—"Substituting Materials in Gear Design", MACHINE DESIGN, Nov., 1942.
31. Foley, Francis B.—"Appraisal of NE Steels in Terms of Old Types", *Metal Progress*, Sept., 1942.
32. Roush, Roy W.—"Evaluating NE Steels", MACHINE DESIGN, Feb., 1943.
33. "National Emergency Steels; NE 1300 Series"—*Metal Progress*, May, 1943.
34. Patton, Walter G.—"Mechanical Properties of NE, SAE and Other Hardened Steels", *Metal Progress*, May, 1943.
35. "Boron"—Comment from Editorial Page of *Metal Progress*, May, 1943, Page 735.
36. Frocht, Max. M.—*Photoelasticity*, Vol. I, John Wiley & Sons, 1941.
37. Carroll, R. H. and J. L. McCloud—"Ford Alloy Castings", *Metals & Alloys*, Aug., 1936.

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37. Carroll, R. H. and J. L. McCloud—"Ford Alloy Castings", *Metals & Alloys*, Aug., 1936.

Substitutions Widen Design Horizons

(*Concluded from Page* 132)

in finding a substitute which, as it turns out, is likely to be permanent. Formerly, an aluminum sand casting was used for the drum in this head. Subsequently a change was made to an aluminum permanent mold casting, with resultant savings in metal and machining time. More recently a switch was made to bakelite, the drum being redesigned with ribs extending down the inside and with a piece of fiberboard at the bottom of a side slot to provide reinforcement. Castellations at the top of the drum were similarly reinforced with interior ribs.

A handwheel on the same company's lathes also has been changed over from an aluminum casting to composition hard rubber reinforced with a steel spider welded to a steel hub machined from bar stock. Development of a plastic with about the same shrinkage factor as composition rubber may make possible a switch to this new material without changing molds. Previously, such a change could not have been made without making new molds.

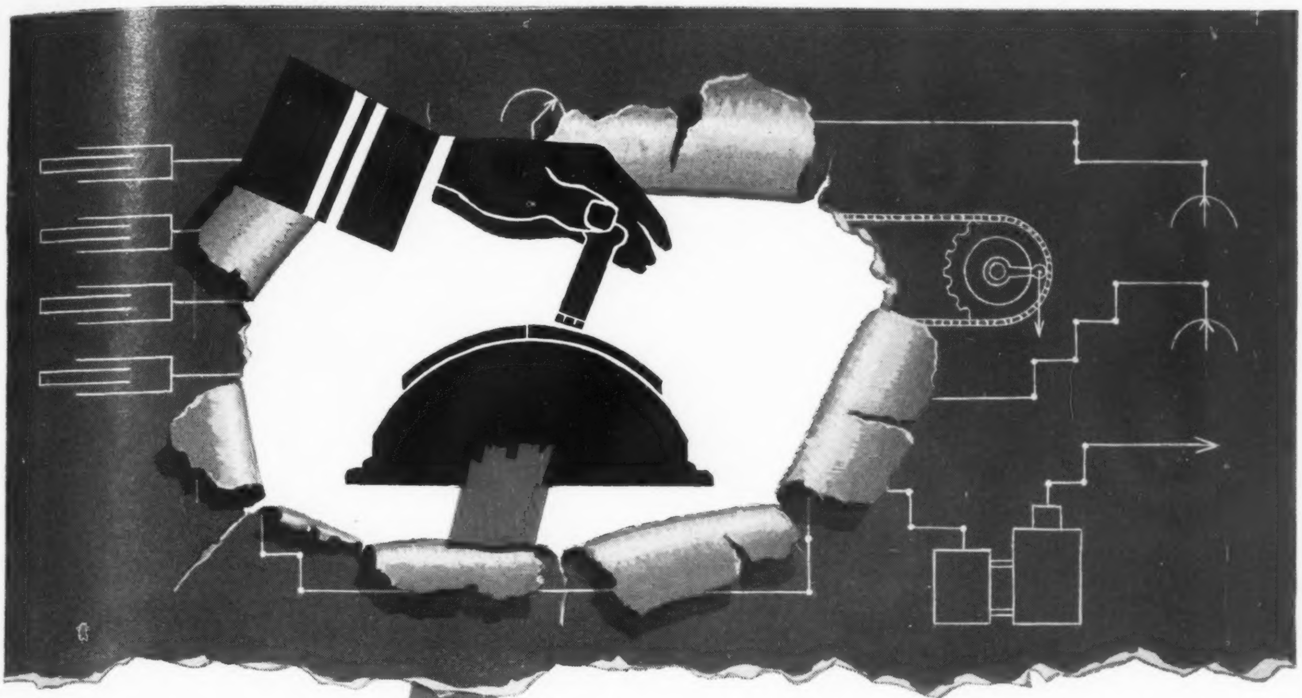
A sample of a type of substitution familiar to most designers is the rotors for the Aridifier made by the Logan Engineering Co., Chicago, for removing condensed moisture, oil and other foreign matter from compressed air and gas lines. These rotors, formerly cast aluminum, now are molded from a phenol formaldehyde compound in one piece with a bronze insert in which the bearings are mounted.

In considering the substitution of plastics for metals, it probably will be found that the type of plastic likely to meet many design requirements is the resin-bonded, laminated composition with superior strength properties which already is being used for such purposes as marine bearings, gears, piston rings and pump valves. *Fig.* 9 illustrates such an application, a fan blade.

Incidentally, considerable significance is attached to the request received by one machine tool builder from a customer in the last few weeks to tool up several machines for turning out screw machine parts from extruded plastic bars. These will be used in place of molded plastic and metal parts.

Sight should not be lost of the possibilities for glass, for which the United States now has an annual productive capacity of some 6,000,000 tons. Containers represent the principal outlet but it also may be had in various bent shapes, polished flat sheets, shatterproof laminations and in the form of fiber for insulation and other purposes.

THERMOPLASTICS for forming light metals are being used in one of California's plants. A large 14,000-pound die for forming an important section of war plane's aluminum-alloy "skin" was recently received, with a rush order for casting a new plastic forming punch to match. By next morning the company had produced a punch from its plastic ready for immediate installation and operation.



BUSINESS AND SALES BRIEFS

FORMED to develop new materials for war uses and after the war, and to extend the wartime usage of present materials and facilities is a new service and development division by Dow Chemical Co. D. K. Ballman will head the new division, which will be headquartered in Midland, Mich. Assisting Mr. Ballman will be a group of experts selected for their technical knowledge of the company's materials and specific industries.

♦

Formerly group leader in charge of research and chemical development, Dr. J. J. Pyle has been appointed chemist in charge of the plastics laboratory at General Electric Co., succeeding Dr. G. F. D'Alelio who resigned from the company. J. W. Underwood will assist Dr. Pyle.

♦

To serve as an adjunct to the Meriden plant of New Departure division, General Motors Corp., a new plant will be operated in Guilford, Conn., for the production of instrument bearings. Francis B. Wasley, who has been in charge of the New York office, will manage the new plant.

♦

George H. Adams, executive vice president and sales director of The Bunting Brass & Bronze Co., announces that William T. Streicher has been placed in charge of bearing sales in the Detroit and Michigan area. Mr. Streicher has heretofore represented the company in Chicago.

♦

Two appointments have been announced by Norma-Hoffmann Bearings Corp., Stamford, Conn.—that of Frank L. Wright, formerly chief metallurgist, to the position of manager of research; and Warren D. Anderson, for over five years a member of the company's engineering staff, to assistant to the chief engineer for the firm. Mr. Wright has been connected with Norma-Hoffmann for eleven years.

♦

James G. Gammel, formerly industrial brush representative in the New England territory for Osborn Mfg. Co., Cleveland, has been appointed technical assistant in brush engineering service at the home office, assisting G. O. Rowland who is chief of the War Production Board's industrial brush unit.

♦

For the past four years Cleveland district representative, John W. Thompson, has been appointed assistant manager of alloy steel sales for Carpenter Steel Co., Reading, Pa. Mr. Thompson will work out of the main office at Reading.

♦

Aircraft parts will be manufactured in the plant addition being made to buildings of Moore Drop Forging Co., Springfield, Mass.

♦

Westinghouse Electric & Mfg. Co. has established a group of electronic engineers who will guide and direct the industrial, central station and transportation electronic applications. This new central organization will supplement the company's present electronic divisions with specialized activity. Centralization is intended to accomplish three primary objectives: First, to expedite present wartime applications of electronics; second,

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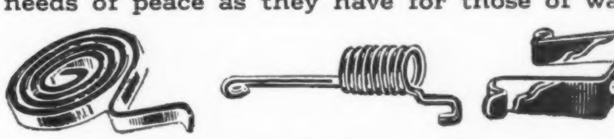
Yes, Victory is our business now—just as it is yours, whoever you are. The better, and the sooner, each Reliable spring fulfills its duty in our vast war program, so much nearer are we all to the day when peace again rules the world. When that time comes, Reliable finer springs stand ready to serve you as faithfully for the needs of peace as they have for those of war.



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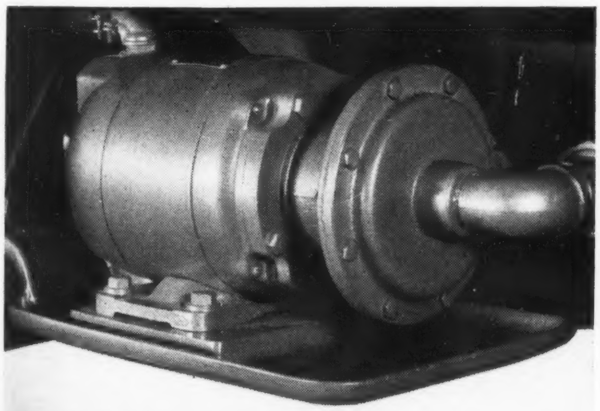

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to be able to utilize to the fullest extent the present development for postwar applications; and third, to make immediately available to all other industries the developments found practical in one industry. Gordon F. Jones and Carl J. Madsen will head the group.

♦

With headquarters at 10345 Linwood avenue, Detroit, The Detroit Electronic Laboratory has been formed for the development and manufacture of special-purpose electronic tubes. Among these tubes is a line designed primarily for control equipment for resistance welding.

♦

Well known as a foundry expert, Dave M. Whyte has been appointed supervisor in charge of casting sales of the Cooper-Bessemer corporation's two foundries, one at the Mt. Vernon, O., headquarters and the other at Grove City, Pa. Mr. Whyte has been connected with the company for twenty years.

♦

An application engineer for Westinghouse in the Spokane, Wash., area since 1935, H. B. Hodgins has been appointed manager of this office. Mr. Hodgins joined the company in 1923 as a graduate student at East Pittsburgh and was soon transferred to the company's San Francisco office where he remained until he started serving the Spokane area in 1935.

♦

For twelve years Detroit representative of Lukens Steel Co., C. T. Hansen has been made Pacific coast sales representative of the company and its subsidiaries, By-Products Steel Corp., and Lukenweld Inc. The C. T. Hansen Co., of which Mr. Hansen is president, has opened offices in Los Angeles, San Francisco and Seattle, with Mr. Hansen in personal charge. Offices are located in Los Angeles 14, at 628 Security Title Insurance building, 530 West Sixth street; in San Francisco 4, at 423-24 Russ building, 235 Montgomery street; and in Seattle 1, at 6115-16-17 Arcade building, 1319 Second avenue.

♦

According to a recent announcement, W. A. Mann has been appointed assistant manager, Industrial division, General Electric's central district, Chicago. His connection with the company goes back to 1923 when he entered the test course there.

♦

Metals Disintegrating Co., Elizabeth, N. J., has announced the appointment of Harold E. Hall as president of the company.

♦

Announced recently was the appointment of Fred J. Wood as district manager of Jessop Steel Co. Ltd., 530 Front street, west, Toronto. Mr. Wood has been connected with the company's Detroit office since 1942.

♦

To be known as the Resin and Insulation Materials division, a new division has been formed by General Electric Co. as part of the Appliance and Merchandise department. With E. L. Feininger as manager, the new division will be responsible for the manufacture, engineering and sales of insulating varnish, glyptal, varnished cloth and mica products.

♦

George M. Muschamp and Paul L. Goldstrohm have been appointed vice presidents of the Brown Instrument Co., Philadelphia. Mr. Muschamp has been with the company in its engineering department for twelve years and in charge of

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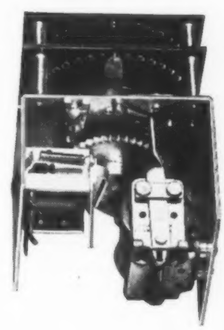

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engineering for the past three years. He becomes vice president in charge of engineering. Mr. Goldstrohm has been connected with the organization for twenty years, and has been assistant general manager for the past two years.

•

Formerly vice president of Bundy Tubing Co. with whom he had been connected for fifteen years, E. Q. Smith has joined the Aga Metal Tube Co., Elizabeth, N. J.

MEETINGS AND EXPOSITIONS

Sept. 6-10—
American Chemical Society. One-hundred-and-sixth meeting to be held at Minneapolis. Charles L. Parsons, Mills building, Washington, is secretary.

Sept. 23-24—
Society of Automotive Engineers Inc. National tractor meeting to be held at the Schroeder hotel, Milwaukee. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Sept. 28-30—
Association of Iron and Steel Engineers. Annual convention to be held at Hotel William Penn, Pittsburgh. Brent Wiley, Empire building, Pittsburgh, is managing director.

Sept. 30-Oct. 2—
Society of Automotive Engineers Inc. National aircraft engineering and production meeting to be held at the Biltmore hotel, Los Angeles. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Oct. 5-7—
National Safety Council. Meeting to be held at Sherman, LaSalle and Morrison hotels, Chicago. W. H. Cameron, 20 North Wacker drive, Chicago, is secretary.

Oct. 13-16—
Electrochemical Society. Meeting to be held at Hotel Pennsylvania, New York. Additional information may be obtained from C. G. Fink, Columbia university, New York.

Oct. 18-22—
Society of Motion Picture Engineers. Meeting to be held in Hollywood. Additional information may be obtained from Sylvan Harris, Hotel Pennsylvania, New York.

Oct. 18-23—
American Welding Society. Twenty-fourth annual meeting to be held at Hotel Morrison, Chicago. M. M. Kelly, 33 West Thirty-ninth street, New York, is secretary.

Oct. 25-29—
National Electric Manufacturers association. Annual meeting to be held at Waldorf-Astoria hotel, New York. Additional information may be obtained from W. J. Donald, 155 East Forty-fourth street, New York.

Nov. 15-16—
American Institute of Chemical Engineers. Meeting to be held in Pittsburgh. Additional information may be obtained from the American Institute of Chemical Engineers, 29 West Thirty-ninth street, New York.

Nov. 29-Dec. 3—
American Society of Mechanical Engineers. Annual meeting to be held in New York. C. E. Davies, 29 West Thirty-ninth street, New York, is secretary.

Dec. 6-11—
Exposition of Chemical Industries. Nineteenth exposition to be held at Madison Square Gardens, New York. Exposition under management of International Exposition Co., Grand Central Palace, New York. Charles F. Roth is president.